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A THERMODYNAMIC ANALYSIS OF GAS ENGINE TESTS

BY

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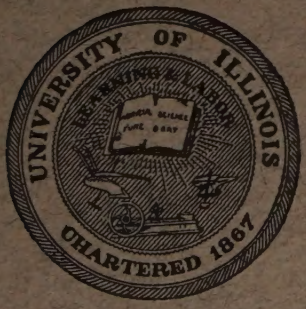

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THE ENGINEERING EXPERIMENT STATION,
UNIVERSITY OF ILLINOIS,
URBANA, ILLINOIS.

UNIVERSITY OF ILLINOIS
ENGINEERING EXPERIMENT STATION

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A THERMODYNAMIC ANALYSIS OF GAS ENGINE TESTS

I. Introduction

1. *Object and Scope of Investigation.*—The object of this investigation was to apply a rational thermodynamic analysis of the constant volume, or Otto, cycle, to test results obtained from an engine operating on such a cycle, and from a comparison of results from theory and experiment to discuss the factors which prevent the actual engine from attaining the ideal performance as defined by the thermodynamic analysis.

Furthermore, by a combination of laboratory data and thermodynamic theory it was possible to throw considerable light on such processes as the rate of reaction in the cylinder of the engine, and the magnitude of the heat losses during different parts of the cycle.

2. *Method of Investigation.*—The thermodynamic theory of this investigation was limited to a discussion of the effects of dissociation, mixture strength, compression ratio, and character of fuel on the performance of the ideal cycle.

The theoretical discussion is based upon a thermodynamic analysis* of the problem of gaseous combustion developed by G. A. Goodenough, Professor of Thermodynamics in the University of Illinois.

The experimental work was done in the Mechanical Engineering Laboratory on an engine running on the constant volume, or Otto, cycle, at constant load and speed, with various compression ratios and mixture ratios, using illuminating gas and hydrogen as fuels.

3. *Acknowledgments.*—The experimental work described in this bulletin was done under the direction of Professor A. C. Willard, Head of the Department of Mechanical Engineering. Professor A. P. Kratz and Professor G. A. Goodenough followed the investigation throughout, and made many suggestions of value in both the theoretical and experimental parts of the problem.

II Thermodynamics of the Ideal Otto Cycle

4. *Definition of Efficiencies.*—A number of different thermal efficiencies will be referred to in the bulletin, and will be defined as follows:

* "An Investigation of the Maximum Temperatures and Pressures Attainable in the Combustion of Gaseous and Liquid Fuels," by G. A. Goodenough and G. T. Felbeck, University of Illinois, Engineering Experiment Station Bulletin 139, 1924.

(a) The ideal thermal efficiency is the ratio of the work obtained from the ideal Otto cycle, computed as shown later, to the work equivalent of the lower heat of combustion* of the fuel used. This ideal thermal efficiency is based on the assumption of complete combustion at the time of maximum pressure in the cycle.

(b) The ideal thermal efficiency, assuming dissociation, is the ratio of the work obtained from the ideal Otto cycle, with incomplete combustion, to the work equivalent of the lower heat of combustion of the fuel used. This efficiency is based on the assumption that dissociation occurs, so that the combustion is not complete at the time of maximum pressure in the cycle.

(c) The thermal efficiency of compression, explosion, and expansion is the ratio of the work equivalent of the area on the actual indicator diagram between the compression, explosion, and expansion lines, to the work equivalent of the lower heat of combustion of the fuel used. This efficiency does not include losses due to suction and exhaust.

(d) The indicated thermal efficiency is the ratio of the work equivalent of the area of the whole indicator diagram to the work equivalent of the lower heat of combustion of the fuel used.

(e) The potential efficiency of compression, explosion, and expansion is the ratio of the thermal efficiency of compression, explosion, and expansion to the ideal thermal efficiency.

(f) The indicated potential efficiency is the ratio of the indicated thermal efficiency to the ideal thermal efficiency.

5. *Requirements of Ideal Cycle.*—When an attempt is made to convert chemical energy into mechanical energy by chemical and thermal processes, a limiting value for the efficiency attainable is immediately established by the nature of the processes chosen. This limiting efficiency, which is called the ideal thermal efficiency, cannot be exceeded as long as the character of the conversion process is not altered. However, it is to be expected that changes in the method of converting energy from one form to another will produce changes in the limiting, or ideal, thermal efficiency. It is therefore necessary, in establishing the ideal thermal efficiency for a given engine, that the properties of the working fluid furnished to the engine be taken

* All heats of combustion used in efficiency calculation are the lower heats of combustion at 60 deg. F.

into consideration as well as the characteristics of the engine itself. For the Otto cycle, the heat of combustion, specific heat, and reaction characteristics of the air-fuel mixture must be considered, as well as the compression ratio, and other physical characteristics of the engine.

6. *Potential Efficiency.*—If an actual engine were to attain the ideal thermal efficiency its performance would be perfect, even though the actual efficiency of conversion of chemical energy into mechanical energy were only 50 per cent. A just measure of the performance of an actual engine is thus the ratio of the indicated thermal efficiency to the ideal thermal efficiency, rather than the actual thermal efficiency itself. This ratio is called the potential efficiency.

7. *Sources of Loss of Efficiency in Actual Engines.*—The difference between the ideal thermal efficiency and the indicated thermal efficiency is caused by (1) mechanical losses and (2) losses due to imperfect operation of the cycle. The mechanical losses are (1) frictional losses and (2) pumping, or induction and exhaust losses. The frictional and pumping losses are not discussed in this bulletin, since they are more or less independent of the thermal processes occurring in the engine. In this study of potential efficiency the mechanical losses are eliminated by determining the indicated thermal efficiency from the indicator card, using the area between the expansion and compression curves as a measure of the work obtained from the cycle. This efficiency is called the indicated efficiency of compression, explosion, and expansion; and the problem is thus reduced to that of determining the maximum efficiency attainable for an engine using a given working fluid, with a known pressure, temperature, volume, and composition within the cylinder at the commencement of the compression stroke.

8. *Determination of Chemical Analysis of Charge within Cylinder.*—The items referred to by numbers in the following calculations are those contained in the tabular computation form, Table 1.

The charge is made up from three sources:

(1) air drawn in,
(2) fuel gas drawn in, and
(3) exhaust gases left in the clearance space.

The method of procedure used in determining the analysis of the charge within the cylinder is illustrated in the following example (p. 19), using the test data from Test No. 22 (see Table 3, pp. 56-59).

TABLE 1

CALCULATION FORM FOR TEST NO. 22

1- Compression Ratio	5.00	19- Exhaust Gas, Mols = (Item 15 / Item 13)	1.446
2- Air-Gas Ratio	5.337	20- Total = M_1	7.783
3- H_t (520°) B.t.u.	182343	21- Volume at 70% Comp. = 0.2848 + Item 12 = V_{10}	0.3865
4- Reciprocal, Item 3	5.48417×10^{-6}	22- Press., lb. per sq. in. abs. = P_{10}	12.3
5- Fuel Gas, cu. ft. per min. at 60° F. and 14.7 lb. per sq. in.	4.227	23- 0.0932642 / Item 16	181.391
6- r.p.m.	337.6	24- $T_{10} = P_{10} \times V_{10} \times$ Item 23, deg. F. abs.	862
7- r.p.m. x 189.55	63992.1	25- $V_{15} = 0.0610 +$ Item 12, cu. ft.	0.1627
8- Air, cu. ft. per min. at 60° F. and 14.7 lb. per sq. in.	22.561	26- Press. lb. per sq. in. abs. = P_{15}	34.7
9- Temp. Exh. Gas, deg. F. abs. = T_4'	1480	27- $P_{15} \times V_{15} \times$ Item 23, $= T_{15}$ deg. F. abs.	1024
10- Press. Exh. Gas, lb. per sq. in. abs. = P_4'	14.9	28- Composition of One Mol of Air: H_2O	0.016
11- Exhaust Gas cu. ft. per Mol, = $\left(\frac{10.7169 \times T_4'}{P_4'}\right)$	1064.5	O_2	0.206
12- Clearance Volume = (0.4068 / Item 1 - 1)	0.1017	N_2	0.778
13- Fuel Gas, Mols per Charge = (Item 5 / Item 7)	0.00006606	28a- $e_o =$ Item 35 + $\frac{K}{2} + \frac{L}{2}$	1.2520
14- Air, Mols per Charge, = (Item 8 / Item 7)	0.00035256	Intermediate Mixture: Kx CO_2	0.763x
15- Exhaust Gas, Mols per Charge = (Item 12 / Item 11)	0.00009554	$K(1-x)$ CO	0.763(1-x)
16- Total Charge, Mols	0.00051416	Ly H_2O	1.315y
17- Fuel Gas per Charge	1.000	$L(1-y)$ H_2	1.315(1-y)
18- Air in Mols (Equals Item 2) = (Item 14 / Item 13)	5.337	$e_o - \frac{K}{2} - \frac{L}{2}$ O_2	1.2520 - 0.3815x - 0.6765y
		r N_2	5.166
		28b- $M_o - \frac{K}{2}x - \frac{L}{2}y$ Total	8.496 - 0.3815x - 0.6765y

	Constituent	Mols	Required Oxygen	H_2O Produced	CO_2 Produced	N_2 Produced
Composition of 1 Mol of Wet Fuel Gas	CO	0.196	0.098	——	0.196	——
	H_2	0.489	0.245	0.489	——	——
	CH_4	0.057	0.114	0.114	0.057	——
	C_2H_2	0.002	0.005	0.002	0.004	——
	C_2H_4	0.106	0.318	0.212	0.212	——
	C_2H_6	0.032	0.112	0.096	0.064	——
	C_6H_6	0.005	0.038	0.015	0.030	——
	CO_2	0.052	——	——	0.052	——
	N_2	0.011	——	——	——	0.011
	H_2O	0.047	——	0.047	——	——
	O_2	0.003	0.003	——	——	——
Air Supplied (Item 2 x Item 28)	H_2O	0.085	——	0.085	——	——
	O_2	1.099	-1.099	——	——	——
	N_2	4.153	——	——	——	4.153
	29- Totals	6.337	29- 0.172	29- 1.060	29- 0.615	29- 4.164

30- Sum = 6.011

Residual Exh. Gas It. 19 x It. 30	31- CO_2	0.148	Products of Complete Combustion of Charge = Item 29 + Item 31	33- $CO_2 = K$	0.763	Item 19 / Item 30 = 0.240588
	31- H_2O	0.255		34- $H_2O = L$	1.315	38- It. 35 + It. 36 = 5.379
	31- O_2	0.041		35- O_2	0.213	$\frac{1}{2}(CO + H_2 + C_2H_2 - C_2H_6 - C_6H_6) =$
	31- N_2	1.002		36- N_2	5.166	0.325 $M_1 - M_2 = 0.326$
32- Total Charge = It. 20 + M_1 = 7.783				37- Total = M_2	7.457	

TABLE 1 (CONTINUED)

CALCULATION FORM FOR TEST NO. 22

32a	Constituent	Mols	T_{15}=1024 H_v per Mol	mH_v at T_{15}	T_o=1400 H_v per Mol	mH_v at T_o	T_{100}=779 H_v per Mol	mH_v at T_{100}
Mol Composition of Charge Within Cylinder	CO	0.196	121 472	23 809	120 976	23 711	121 625	23 838
	H_2	0.489	103 331	50 529	103 612	50 666	103 160	50 445
	CH_4	0.057	345 463	19 691	346 304	19 739	345 461	19 691
	C_2H_2	0.002	542 158	1 084	542 979	1 085	541 841	1 084
	C_2H_4	0.106	576 657	60 808	572 534	60 689	574 470	60 894
	C_2H_6	0.032	608 109	19 459	606 436	19 406	609 381	19 500
	C_6H_6	0.005	1 359 010	6 795	1 360 906	6 804	1 359 000	6 795
	CO_2	0.200	38a – 182 175 H_v at T_{15}		39 – 182 100 H_v at T_o		40 – 182 247 H_v at T_{100}	
	H_2O	0.387						
	O_2	1.143						
	N_2	5.166						
32 – Total		7.783	41 – Diatomic = $CO+O_2+N_2$ = 6.505					

41a	Constituent	Mols m	u at T_{10} T_{10}=862°	mu_{10}	u at T_{15} T_{15}=1024	mu_{15}
Initial Charge Within Cylinder	CO_2	0.200	5772	1154	7119	1424
	H_2O	0.387	5458	2112	6504	2517
	Diatomic	6.505	4288	27893	5106	33215
	H_2	0.489	3708	1813	4461	2181
	CH_4	0.057	5194	296	7046	402
	C_2H_2	0.002	6634	13	8553	17
	C_2H_4	0.106	6565	696	8363	886
	C_2H_6	0.032	7604	243	9747	312
	C_6H_6	0.005	13552	68	18736	94
	Total	7.783	42-Total	34288	43-Total	41048

Work 15% Comp to x% on Exp.		
x%	Area, sq.in.	49-B.t.u. = W
2.5	– 1.922	-4381
5	+ 0.060	+ 137
7.5	1.974	4500
10	3.747	8541
15	6.846	15605
20	9.480	21610
30	13.781	31414
40	17.188	39180
50	19.986	45558
60	22.268	50760
70	24.252	55282
80	25.964	59185
90	27.467	62611
100		

44 – Work, B.t.u. per sq. in. = 0.0150589 × Item 20 / Item 16	2279.5
45 – Work Area 70% to 15% Comp.	5.555 sq.in.
46 – Compression Work = Item 44 × Item 45,	12 663 B.t.u.
47 – Energy Increase upon Comp. 70% to 15% = Item 43 – Item 42,	6760 B.t.u.

50 – Work of Exp. – Work of Comp. (By Planimeter): 21.92 | 49967

48 – Heat Loss During Comp. 70% to 15% = Item 46 – Item 47 =	5903
In Per Cent of Hv_{520}	3.24

51 – $T = \frac{PV}{MR} = PV \times$ Item 23 $\times \frac{\text{Item 32}}{\text{Item 37}} = PV \times 189.321$

TABLE 1 (CONTINUED)

CALCULATION FORM FOR TEST NO. 22

52- % of Expansion Stroke		2.5	5	7.5	10	15	20	30	40	50	60	70	80	90
53- Piston Displacement		0.0102	0.0203	0.0305	0.0407	0.0610	0.0814	0.1220	0.1627	0.2034	0.2441	0.2848	0.3254	0.3661
54- Item 53+ Item 12 = V, cu.ft.		0.119	0.1220	0.1322	0.1424	0.1627	0.1831	0.2237	0.2644	0.3051	0.3458	0.3865	0.4271	0.4678
55- Pressure, P, lb. per sq. in., from Indicator Diagram		159.0	156.8	148.5	135.4	113.8	97.8	76.0	61.2	50.6	42.5	36.6	32.0	28.1
56- PV		17.792	19.130	19.632	19.281	18.515	17.907	17.001	16.181	15.438	14.697	14.146	13.667	13.145
57- T. = Item 51	T_5 = 1024	3368	3622	3717	3650	3505	3390	3219	3063	2923	2782	2678	2587	2489
U_{CO_2}	7119	31894	34861	35977	35190	33491	32150	30167	28368	26763	25155	23975	22949	21852
U_{H_2O}	6504	25192	27871	28917	28176	26613	25417	23697	22192	20889	19620	18711	17932	17112
$U_{diat.}$	5106	18183	19812	20434	19995	19054	18322	17252	16295	15455	14619	14011	13486	12925
KU_{CO_2} = 0.763 X U_{CO_2}	5432	24335	26599	27450	26850	25554	24530	23017	21645	20420	19193	18293	17510	16673
LU_{H_2O} = 1.315 X U_{H_2O}	8553	33127	36650	38026	37051	34996	33423	31162	29182	27469	25800	24605	23581	22502
Item 38 X U_D = 5.319 X U_D	27465	97806	106569	109915	107553	102491	98554	92799	87651	83132	78636	75365	72541	69524
58- $\Sigma mu = U$	41450	155268	169818	175391	171454	163041	156507	146978	138478	131021	123639	118263	113632	108699
59- ΔU		113818	118368	133941	130004	121591	115057	105528	97028	89571	82189	76813	72182	67249
61- Item 38a - Item 59 - Item 49		72738	53670	43734	43630	44979	45508	45233	45967	47046	49236	50080	50808	52315
62- $H+(1-z)H_v$ = Item 61 X Item 4 = % $H_{v\,520°}$		39.89	29.43	23.98	23.93	24.67	24.96	24.76	25.22	25.80	27.00	27.46	27.86	28.69

TABLE 1 (CONTINUED)

CALCULATION FORM FOR TEST NO. 22

T		T_{70}=862	700	800	1300	1400	1500
	ϕ_{CO_2}	38.0505	36.4195	37.4543	41.5968	42.2884	42.9479
	ϕ_{H_2O}	42.8069	41.4773	42.3287	45.4933	45.8930	46.4645
	$\phi_{Diat.}$	33.4694	32.4249	33.0941	35.5580	35.9407	36.2993
	ϕ_{H_2}	27.7131	26.7693	27.3722	29.6548	30.0190	30.3626
	ϕ_{CH_4}	19.0658	17.0483	18.3011	24.2967	25.4620	26.6196
	$\phi_{C_2H_2}$	35.4049	33.2174	34.5889	40.6804	41.8021	42.9022
	$\phi_{C_2H_4}$	37.5289	35.4521	36.7577	42.4323	43.4595	44.4627
	$\phi_{C_2H_6}$	41.9890	39.5290	41.0719	47.8554	49.0945	50.3074
	$\phi_{C_6H_6}$	41.0315	35.4604	38.9094	55.7878	59.1170	62.4361
m=Item 41a		862	700	800	1300	1400	1500
0.200	$m\phi_{CO_2}$	7.6101	7.2839	7.4909	8.3194	8.4577	8.5896
0.387	$m\phi_{H_2O}$	16.5663	16.0517	16.3812	17.6059	17.7993	17.9818
6.505	$m\phi_{Diat.}$	217.7184	210.9240	215.2771	231.3048	233.7942	236.1269
0.489	$m\phi_{H_2}$	13.5517	13.0902	13.3850	14.5012	14.6793	14.8473
0.057	$m\phi_{CH_4}$	1.0868	0.9717	1.0432	1.3849	1.4513	1.5173
0.002	$m\phi_{C_2H_2}$	0.0708	0.0664	0.0692	0.0814	0.0836	0.0858
0.106	$m\phi_{C_2H_4}$	3.9780	3.7579	3.8963	4.4978	4.6067	4.7130
0.032	$m\phi_{C_2H_6}$	1.3429	1.2649	1.3143	1.5314	1.5710	1.6098
0.005	$m\phi_{C_6H_6}$	0.2051	0.1773	0.1945	0.2789	0.2956	0.3122
Total = $\Sigma m\phi$		262.1301	253.5880	259.0517	279.5057	282.7387	285.7837
64- $\Delta\phi$			-8.5421	-3.0784	17.3755	20.6086	23.6536
65- First Diff.			5.4637		3.2331	3.0450	

V_{70} = Item 21	0.3865		
V_o = Item 12	0.1017	V_{100} = 0.04068 + Item 12	0.5085
65a- $\frac{V_{70}}{V_o}$	3.8	67a- $\frac{V_{100}}{V_{70}}$	1.3156
$Log_{10}\frac{V_{70}}{V_o}$	0.57978	$-Log_{10}\frac{V_{100}}{V_{70}}$	-0.11916
M_1 = Item 20	7.783		
66- 4.571 X M_1	35.5761		
67- 4.571 X M_1 X $Log_{10}\frac{V_{70}}{V_o}$	20.6263	68- 4.571 X M_1 X $Log_{10}\frac{V_{100}}{V_{70}}$	-4.2392
Item 64	20.6086	Item 64	-3.0784
Item 67 - Item 64	0.0177	Item 68 - Item 64	-1.1608
Item 65	3.2331	Item 65	5.4637
$\frac{\text{It. 67} - \text{It. 64}}{\text{It. 65}}$ X 100	0+	$\frac{\text{It. 68} - \text{It. 64}}{\text{It. 65}}$ X 100	-21
69- T_o in deg. F. abs.	1400	71- T_{100} in deg. F. abs.	779
T_{70} in deg. F. abs. = It. 24	862		
P_{70} in lb. per sq. in. abs. = It. 22	12.3		
$\frac{P_{70}}{T_{70}}$	0.0142691		
T_o X $\frac{V_{70}}{V_o}$ = Item 69 X Item 65a	5320	T_{100} X $\frac{V_{70}}{V_{100}} = \frac{\text{Item 71}}{\text{Item 67a}}$	592.125
70- $P_o = T_o$ X $\frac{V_{70}}{V_o}$ X $\frac{P_{70}}{T_{70}}$	75.91	72- $P_{100} = T_{100}$ X $\frac{V_{70}}{V_{100}}$ X $\frac{P_{70}}{T_{70}}$	8.45

TABLE 1 (CONTINUED)

CALCULATION FORM FOR TEST NO. 22

T		T_o 1400	4600	4700	4800
U_{CO_2}		10504	46536	47752	48973
U_{H_2O}		9000	39992	41412	42870
$U_{Diat.}$		7033	26641	27394	28159
m	mu				
lt. 33, 0.763	mu_{CO_2}	8014	35507	36435	37366
lt. 34, 1.315	mu_{H_2O}	11835	52589	54457	56374
lt. 38, 5.379	$mu_{Diat.}$	37831	143302	147352	151467
73a- Sum = U_{P_o}		57680			
73- ΔU_P			173718	180564	187527
74- 1st Diff.			6846	6963	

Item 39 = H_v at T_o	182100
Item 73	180564
Item 39 - Item 73	1536
$\frac{\text{lt. 39 - lt. 73}}{\text{Item 74}} \times 100$	22
75- T_e (Complete Combustion)	4722

P_o = Item 70	75.91
M_1 = Item 20	7.783
$M_1 T_o$	10896.2
76- $\frac{P_o}{M_1 T_o}$	0.00696665
M_2 = Item 37	7.457
$M_2 T_e$	35212
77- $P_e = M_2 T_e \times \frac{P_o}{M_1 T_o}$	245.31
77a- $\frac{V_4}{V_3}$ = Item 1	5.00
Log $\frac{V_4}{V_3}$	0.69897
77b- 4.571 × M_2	34.0859
78- 4.571 M_2 Log $\frac{V_4}{V_3}$	23.8250

T		T_o 4722	3100	3200	3300
ϕ_{CO_2}		55.7077	50.7292	51.0954	51.4516
ϕ_{H_2O}		57.0915	52.1857	52.4928	52.7979
$\phi_{Diat.}$		43.1728	40.3306	40.5254	40.7166
m	$m\phi$				
lt. 33, 0.763	$m\phi_{CO_2}$	42.5050	38.7064	38.9858	39.2576
lt. 34, 1.315	$m\phi_{H_2O}$	75.0753	68.6242	69.0280	69.4292
lt. 38, 5.379	$m\phi_{Diat.}$	232.2265	216.9383	217.9861	219.0146
$\phi = \Sigma m\phi$		349.8068			
79 $\Delta\phi$			25.5379	23.8069	22.1054
80 1st. Diff.				1.7310	1.7015

Item 78	23.8250
Item 79	25.5379
Item 78 - Item 79	1.7129
Item 80	1.7310
$\frac{\text{Item 78 - Item 79}}{\text{Item 80}} \times 100$	99
81- T_4 in deg. F. abs.	3199

Item 37 = M_2	7.457
$M_2 T_4$	23854.9
Item 76	0.00696665
$M_2 T_4$ × Item 76	166.188
82- $P_4 = \frac{M_2 T_4 \times \text{Item 76}}{\text{Item 77a}}$	33.24
in lb. per sq. in.	

T		T_o = lt. 71 779	T_4 = lt. 81 3199	T_e = lt. 118 3209
U_{CO_2}		5112	29936	30052
U_{H_2O}		4926	23500	23598
$U_{Diat.}$		3871	17128	17190
m	mu			
lt. 33, 0.763	mu_{CO_2}	3900	22841	22930
lt. 34, 1.315	mu_{H_2O}	6478	30902	31031
lt. 38, 5.379	$mu_{Diat.}$	20822	92131	92465
83b- U_P		31200	145874	146426
$U_{P_4} - U_{P_1}$			83 114674	83a 115226

84- Item 40 = H_v at T_{100}	182247
85- Item 40 - Item 83	67573
86- Complete Combustion, Ideal Efficiency, lt. 4 × lt. 85	37.06%
87- Efficiency of Compression Explosion, and Expansion; lt. 4 × lt. 50	27.40%
87a- Item 40 × Item 83a	67021
87b- Ideal Efficiency with Dissociation = Item 87a × Item 4	36.76%

TABLE 1 (CONTINUED)

CALCULATION FORM FOR TEST NO. 22

T	4550	4600	4650
$U_{Diat.}$	26267	26641	27017
U_{H_2}	25169	25522	25877
$(K+e_o+r) \times U_{Diat.}$	188623	191309	194009
LU_{H_2}	33097	33561	34028
89- Sum + Item 88	209419	212569	215736
H_{Vco} at T	114406	114358	114314
90- KH_{Vco}	87292	87255	87222
$H_{V_{H_2}}$ at T	101830	101673	101510
91- $LH_{V_{H_2}}$	133906	133700	133486
$a = \frac{\text{Item 90}}{\text{Item 91}}$	0.6519	0.6526	0.6534
$b = \frac{\text{Item 89}}{\text{Item 91}}$	1.5639	1.5899	1.6162
c	6.7744	6.8781	6.9820
$c(b-1)$	3.8201	4.0574	4.3023
92- $c(b-1)-(a+b)$	1.6043	1.8149	2.0327
93- $(\text{Item } 92)^2$	2.5738	3.2939	4.1319
$a(c-1)$	3.7643	3.8360	3.9086
$4b$	6.2556	6.3596	6.4648
94- $4ba(c-1)$	23.5480	24.3954	25.2683
95- Item 94 + Item 93	26.1218	27.6893	29.4002
96- $\sqrt{\text{Item } 95}$	5.1109	5.2621	5.4222
97- Item 96 + Item 92	6.7152	7.0770	7.4549
98- $2a(c-1)$	7.5286	7.6720	7.8172
99- $x = \frac{\text{Item 97}}{\text{Item 98}}$	0.8920	0.9224	0.9537
$1-x$	0.1080	0.0776	0.0463
ax	0.5815	0.6020	0.6231
100- $y = b - ax$	0.9824	0.9879	0.9931
Item 28a = e_o	1.2520		
$\frac{K}{2}x$	0.3403	0.3519	0.3638
$\frac{1}{2}y$	0.6459	0.6495	0.6530
e''	0.2658	0.2506	0.2352
$\text{Log } e''$	$\bar{1}.4245$	$\bar{1}.3990$	$\bar{1}.3714$
$\text{Log } x$	$\bar{1}.9504$	$\bar{1}.9649$	$\bar{1}.9794$
$\frac{1}{2}\log \frac{M_1 T_o \times 14.7}{P_o} = \text{It. } 106$	1.6621	1.6621	1.6621
$\text{Log}(1-x)$	$\bar{1}.0334$	$\bar{2}.8898$	$\bar{2}.6656$
$\frac{1}{2}\text{Log } e''$	$\bar{1}.7122$	$\bar{1}.6995$	1.6857
107- $R(xy) = \log x + \frac{1}{2}\log \frac{M_1 T_o \times 14.7}{P_o} - \text{Log}(1-x) - \frac{1}{2}\log e'' =$	2.8669	3.0377	3.2902
$\text{Log } T$	3.6580	3.6628	3.6674
$\frac{1}{2}\text{Log } T$	1.8290	1.8314	1.8337
$\text{Log } K_{P_{co}}$	1.3231	1.2609	1.2001
108- $L(T) = \frac{1}{2}\text{Log } T + \text{Log } K_{P_{co}}$	3.1521	3.0923	3.0338

Item 69 = T_o	1400
H_{Vco} at T_o	120976
$H_{V_{H_2}}$ at T_o	103612
U_{Diat} at T_o	7033
U_{H_2} at T_o	6275
K	0.763
L	1.315
$K+e_o+r$	7.181
KH_{Vco}	+ 92305
$LH_{V_{H_2}}$	+136250
Item 39, Total H_V at T_o	-182100
$(K+e_o+r) \times U_{Diat.}$	- 50504
LU_{H_2}	- 8252
88- Sum	-12301

101- With Dissoc., T_e''	4610
102- x	0.9288
103- y	0.9888
Item 28b, M_o	8.4960
103a- $\frac{K}{2}x$	-0.3543
103b- $\frac{1}{2}y$	-0.6501
104- M''	7.4916
$M''T_e''$	34536.3
Item 76, = $\frac{P_o}{M_1 T_o}$	0.00696665
105- $P_e'' = M''T_e'' \times$ It. 76	240.6
Log 14.7	1.1673
Log Item 76	$\bar{3}.8430$
Log 14.7 - Log It. 76	3.3243
106- $\frac{1}{2}\text{Log } \frac{M_1 T_o \times 14.7}{P_o}$	1.6621

TABLE 1 (CONCLUDED)

CALCULATION FORM FOR TEST NO. 22

T	4610		Log Ly	0.1140	0.1189
x	0.9288	1.000		$\bar{2}$.1673	∞
y	0.9888	1.000	Log L(1-y)	−1.8327	∞
M	Item 104 7.4916	Item 37 7.457	Log e"	$\bar{1}$.3938	$\bar{1}$.3284
		(V x lt.1)		−0.6062	−0.6716
$\frac{10.722}{\text{Item 76}} = V$, cu. ft.	1539.1	7695.5			
			Kx Log Kx	−0.1060	−0.0897
109- Kx	0.7087	0.763	K(1-x) Log[K(1-x)]	−0.0687	0.0
K(1-x) = K - lt. 109	0.0543	0.0	Ly Log Ly	+0.1482	+0.1563
			L(1-x) Log[L(1-y)]	−0.0269	0.0
110- Ly	1.3003	1.315	e" Log e"	−0.1501	−0.1431
L(1-y) = L - lt. 110	0.0147	0.0			
			111- Sum	−0.2035	−0.0765
Item 28a = e_o	1.2520				
− Item 103a	0.3543		4.571 x Item 111	112 −0.9302	113 −0.3497
− Item 103b	0.6501				
		Item 35	$\frac{1544}{V}$	1.00318	0.20064
e"	0.2476	0.2130			
			Log $\frac{1544}{V}$	0.00137	$\bar{1}$.30241
Log Kx	$\bar{1}$.8504	$\bar{1}$.8825			lt. 77b
	−0.1496	−0.1175	4.571 x M	34.2441	34.0859
	$\bar{2}$.7348	∞			
Log K(1-x)	−1.2652	∞	114- 4.571 M Log $\frac{1544}{V}$	+0.0469	-23.7780

T	4610	3100	3200	3300
α	19.3170	20.0119	19.9381	19.8686
β	12.7263	13.3593	13.3437	13.3244
$\phi_{Diat.}$	42.9915	40.3306	40.5254	40.7166
ϕ_{H_2}	36.9440	34.3440	34.5382	34.7285
− Item 109 x α = −Kxα	−13.6899	−15.2691	−15.2128	−15.1597
− Item 110 x β = −Lyβ	−16.5480	−17.5675	−17.5470	−17.5216
(K + e_o + r) x $\phi_{Diat.}$	308.7220	289.6140	291.0129	292.3859
L ϕ_{H_2}	48.5814	45.1624	45.4177	45.6680
− Item 112 or 113	(112) 0.9302	(113) 0.3497	(113) 0.3497	(113) 0.3497
− Item 114	− 0.0469	23.7780	23.7780	23.7780
S	115 327.9488	116 326.0675	116 327.7985	116 329.5003
117- 1st. Diff.		1.7310	1.7018	

Item 115	327.9488	Item 37 = M_2	7.457
Item 116	327.7985	M_2T_4 = Item 37 x Item 118	23929.5
Item 115 − Item 116	0.1503	Item 76	0.00696665
$\frac{\text{Item 115 - Item 116}}{\text{Item 117}} \times 100$	9	119- M_2T_4 x Item 76	166.708
118-(With Dissoc.), T_4	3209° F. abs.	120- P_4 = $\frac{\text{Item 119}}{\text{Item 1}}$ (With Dissoc.)	33.34 lb. per sq. in. abs.

Engine Data:

Engine speed	337.6	r. p. m.
Suction strokes	168.8	per min.
Clearance volume	0.1017	cu. ft.

Air Drawn into Cylinder:

Volumo of air	22.561	cu. ft. per min. at 60 deg. F. and 14.7 lb. per sq. in.

Analysis of air:

H_2O	1.6	per cent
O_2	20.6	per cent
N_2	77.8	per cent

Volume of one mol of any gas at 60 deg. F. abs. and 14.7 lb. per sq. in.: 379.1 cu. ft.

Mols of air per charge:

$$\frac{22.561}{168.8} \times \frac{1}{379.1} = 0.00035256 \text{ (Item 14)}$$

Fuel Gas Drawn into Cylinder:

Volume of fuel gas	4.227	cu. ft. per. min. at 60 deg. F. and 14.7 lb. per sq. in.

Mols of fuel gas per charge:

$$\frac{4.227}{168.8} \times \frac{1}{379.1} = 0.00006606 \text{ (Item 13)}$$

Exhaust Gases Left in Clearance Space:

Temperature of exhaust gases	1480.0	deg. F.*
Pressure of exhaust gases	14.9	lb. per. sq. in.*

Volume of one mol of gas at 1480 deg. F. abs. and 14.9 lb. per sq. in:

$$\frac{1480}{520} \times \frac{14.7}{14.9} \times 379.1 = 1064.5 \text{ cu. ft.}$$

Mols of exhaust gases left in clearance space:

$$\frac{0.1017}{1064.5} = 0.00009554 \qquad \text{(Item 15)}$$

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* The exhaust valve closed on dead center. The temperature given is that of the gas issuing from the valve just before its closing. The pressure is that measured on dead center, as shown on the low pressure indicator card (see Fig. 1).

Collecting the results, the composition of the charge within the cylinder is as follows:

	Mols per Stroke	Mols per Mol of Fuel Gas	
Fuel gas	0.000 066 06	1.000	
Air	0.000 352 56	5.337	
Exhaust gas	0.000 095 54	1.446	
Total	0.000 514 16	7.783	(Item 20)

The second column gives the composition based on the amount of fuel gas as a unit.

The chemical analysis of the exhaust gas is assumed to be that of the products of combustion of the mixture of fuel gas and air undiluted by exhaust gas. This is an approximation, since the actual exhaust gas is the result of the combustion of a diluted mixture. By a progressive calculation the analysis of the products of combustion of the diluted charge can be determined. However, the difference between the products of combustion from the diluted and undiluted charges is less than the probable experimental error in the fuel gas analysis, and on this basis the approximation is justified.

Items 32 to 37 give the computation of the analysis of the original charge and of the products of combustion.

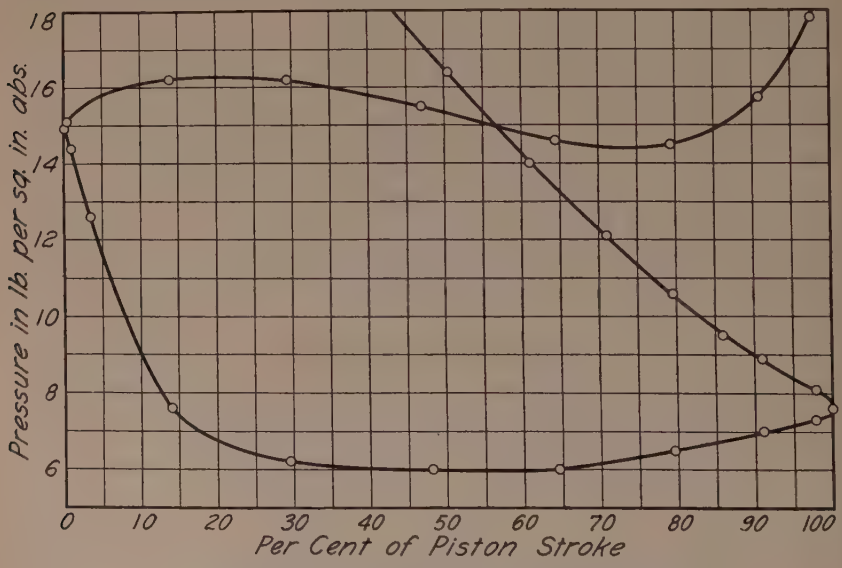

FIG. 1. LOW PRESSURE INDICATOR DIAGRAM FOR TEST NO. 22

9. *Calculation of Initial Temperature.*—In order to begin the computation of the ideal cycle it is necessary to know the temperature, pressure, volume, weight, and composition of the charge within the cylinder at some point during the cycle. The following method of computation gives the physical properties of the charge at a point 70 per cent of the piston stroke from the head end of the cylinder, on the compression stroke. This point is used as a starting point for the computation of the ideal cycle.

Since the pressure, volume, and weight of the charge within the cylinder are known, the temperature at any point on the compression curve previous to ignition can be determined by the use of the perfect gas law $PV = MRT$. The pressure and volume at the end of the suction stroke cannot be used to determine the initial temperature since the inlet valve is still open and gases are flowing into the cylinder. Also the charge is not as yet thoroughly mixed. To allow ample time for mixing and to be certain that the complete charge is within the cylinder, the calculations are based on conditions at 70 per cent of the piston stroke from the head end. The initial temperature to be used as a starting point for the computation of the ideal cycle is calculated as follows (Items 21 to 24):

Pressure at 70 per cent stroke $= P_{70} = 12.3$ lb. per sq. in. abs.

Volume = clearance + 70 per cent piston displacement =
$0.1017 + 0.70 \times 0.4068 = 0.3865$ cu. ft.

$M = 0.000\ 514\ 16$ mols (p. 20)

$$T = \frac{PV}{MR} = \frac{12.3 \times 144 \times 0.3865}{0.000\ 514\ 16 \times 1544} = 862 \text{ deg. F. abs. (Item 24)}$$

10. *Equation for Adiabatic Compression.*—Having established the point at 70 per cent compression stroke as a starting point, the gas is compressed adiabatically. For the purposes of computation it has been found convenient to use a certain function ϕ. The development of this function follows.

From the first law of thermodynamics

$$dq = du + pdv \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where dq = heat added
du = change of thermal energy
pdv = pressure multiplied by change of volume, or the work done, expressed in B.t.u.

For the adiabatic case $dq = 0$. Therefore

$$du = -pdv \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1a)$$

also

$$du = \gamma_v dT$$

where γ_v is the instantaneous specific heat at constant volume of the whole charge being compressed. The specific heat may be expressed as a function of the temperature

$$\gamma_v = a + bT + fT^2$$

From the perfect gas law

$$P = \frac{MRT}{V}$$

Equation (1) may now be written

$$\left(\frac{a}{T} + b + fT\right)dT = -MR\,\frac{dv}{V} \quad . \quad . \quad . \quad . \quad (1b)$$

Integrating between the limits T_1 and T

$$a \log_e \frac{T}{T_1} + b\,(T - T_1) + \tfrac{1}{2} f(T^2 - T_1^2) = MR \log_e \frac{V_1}{V} \quad . \quad (1c)$$

where the subscript 1 denotes the initial state. Rearranging, the equation for adiabatic compression may be written

$$\phi\,(T) - \phi\,(T_1) = MR \log_e \frac{V_1}{V} \quad . \quad . \quad . \quad (2)$$

or, if m is the number of mols of each constituent in the charge, they can be summed up as follows:

$$\Sigma m\phi - \Sigma m\phi_1 = MR \log_e \frac{V_1}{V}$$

Changing* to common logarithms

$$\Sigma m\phi - \Sigma m\phi_1 = 4.571\,M \log \frac{V_1}{V} \quad . \quad (2a)$$

11. *Calculation of Temperature at End of Adiabatic Compression.*—The equation for ϕ developed in Section 10 is used to determine the temperature at the end of adiabatic compression. Since V and V_1 are known and $\phi(T_1)$ can be computed from the initial conditions, $\phi(T)$ can be calculated. The value of $\phi(T)$ as calculated above is then compared with the values of $\phi(T)$ for several assumed

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* $R = 1.985$
Logarithmic conversion factor = 2.30259
1.985 x 2.30259 = 4.571

temperatures (as calculated from the table of ϕ), and by interpolation the final temperature is determined.

The volume at 70 per cent of piston stroke is

$$0.70 \times 0.4068 + 0.1017 = 0.3865 \text{ cu. ft. (Item 21)}$$

The volume at the end of compression is the clearance volume, or 0.1017 cu. ft. (Item 12).

Then

$$\frac{V_1}{V} = \frac{V_{70}}{V_0} = \frac{0.3865}{0.1017} = 3.8$$

The mass of the charge is 7.783 mols (Item 20).

Then

$$\Sigma m\phi - \Sigma m\phi_1 = 4.571 \times 7.783 \log_{10} 3.8$$
$$= 20.6263 \text{ (Item 67)}$$

The values of the ϕ functions have been computed from equation (1c) for the various gases and tabulated for 10-degree intervals. Taking the value of ϕ for each constituent of the charge at 862 deg. F. abs. (the temperature at 70 per cent stroke) and multiplying by the weight of the constituent, the sum is

$$\Sigma m\phi_1 = 262.1301 \text{ (see p. 15, Table 1).}$$

Then

$$\Sigma m\phi = \Sigma m\phi_1 + 20.6263$$
$$= 262.1301 + 20.6263$$
$$= 282.7564$$

Assuming the temperature at the end of adiabatic compression to be 1300 deg., 1400 deg., or 1500 deg. F. abs., the values of $\Sigma m\phi$ are computed, giving

at 1300 deg. . . . . $\Sigma m\phi = 279.5057$
1400 deg. . . . . $\Sigma m\phi = 282.7387$
1500 deg. . . . . $\Sigma m\phi = 285.7837$

By interpolation between these values T_0, at the end of adiabatic compression, is found to be 1400 deg. F. abs. (Items 64 to 69).

Computing the pressure at the end of adiabatic compression

$$P_0 = P_{70} \times \frac{T_0}{T_{70}} \times \frac{V_{70}}{V_0} = 12.3 \times \frac{1400}{862} \times \frac{0.3865}{0.1017}$$

$$= 75.91 \text{ lb. per sq. in. abs. (Item 70)}$$

12. *Calculation of Temperature at Beginning of Adiabatic Compression.*—By the same method as used in the preceding paragraph

the temperature and pressure at the beginning of adiabatic compression are computed, giving

$$T_{100} = 779 \text{ deg. F. abs. (Item 71)}$$
$$P_{100} = 8.45 \text{ lb. per sq. in. abs. (Item 72)}$$

13. *Calculation of Maximum Temperature of Complete Combustion.*—For the combustion of the compressed charge at constant volume equation (1) reduces to

$$du = dq \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

where q is the heat developed by the reaction carried to completion, or H_v at 1400 deg., the compression temperature. This heating value is computed in Item 39,* and is found to be 182 100 B.t.u. per mol.

From equation (3), if $dq = H_v$, the heat of combustion at constant volume

$$H_v = U_{p_e} - U_{p_0} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

where U_{p_e} is the energy of the products mixture at the maximum temperature and U_{p_0} is the energy at the compression temperature. From Goodenough and Felbeck's tables of thermal energy† the values of u for each constituent are found for a temperature of 1400 deg. F. abs. The total energy U_{p_0} is thus 57 680 B.t.u. (Item 73a). Assuming the temperature of complete combustion to be from 4600 to 4800 deg. F. abs., the energies at those temperatures are computed. By comparison of the value of H_v with the difference $U_{p_e} - U_{p_0}$, as given in Item 73, the maximum temperature of complete combustion is found to be 4722 deg. F. abs. (Item 75).

Computing the pressure at the end of the complete adiabatic combustion at constant volume

$$P_c = \frac{M_2}{M_1} \times \frac{T_e}{T_0} \times P_0 = \frac{7.457}{7.783} \times \frac{4722}{1400} \times 75.91$$
$$= 245.31 \text{ lb. per sq. in. abs.}$$

M_1 and M_2 are the weight of the charge in mols before and after complete combustion (Items 20 and 37).

14. *Calculation of Temperature at End of Adiabatic Expansion.*—Using the ϕ functions in the same way as outlined in Section 11, the volume ratio being

$$\frac{V_{100}}{V_0} = 5.00 \text{ (the compression ratio)}$$

* The heating values of the various constituents of the charge are those given in Tables 34 to 36 of Eng. Exp. Sta. Bul. No. 139.

† Eng. Exp. Sta. Bul. No. 139 Tables 25 to 29.

the temperature T_4 at the end of adiabatic expansion from 4700 deg. F. abs. is found to be 3199 deg. F. abs. (Items 79 to 81).

The pressure P_4 at this point is found to be 33.24 lb. per sq. in. abs. (Item 82).

15. *Calculation of Maximum Temperature of Incomplete Combustion.*—The following calculations are based on the assumption that the whole mixture within the cylinder is in a state of chemical equilibrium at the instant of maximum temperature. The complete development of the theory and methods of calculation is contained in Bulletin 139 of the University of Illinois Engineering Experiment Station.

The general method consists of the graphical determination of the three variables T, x, and y, where T is the absolute temperature at which chemical equilibrium occurs, x is the degree of completion of the reaction $CO + \frac{1}{2} O_2 \rightleftarrows CO_2$, and y is the degree of completion of the reaction $H_2 + \frac{1}{2} O_2 \leftrightarrows H_2O$. For a complete reaction, x and y are both unity. For the determination of these three variables three equations are necessary.

The first of the three required equations is the "energy equation," which expresses the relation between the thermal and chemical energies of the mixture before ignition and at chemical equilibrium. This equation is derived from a consideration of the thermal and chemical energies of the working fluid before ignition and at the time equilibrium is attained.*

In its final form it is

$$LyH_v''_{H_2} = -H'_v + KH_{v_{CO}} + LH_{v_{H_2}} + L\triangle u_{H_2} + (K + e_0 + r)\triangle u_D - Kx\,H''_{v_{CO}} \quad \cdots \quad (5)$$

in which†

L = mols of H_2 in the initial mixture
K = mols of CO in the initial mixture
x = proportion of CO burned to CO_2
y = proportion of H_2 burned to H_2O
$H''_{v_{H_2}}$ = heat of complete combustion of H_2 } at final temperature T_e
$H''_{v_{CO}}$ = heat of complete combustion of CO }
H'_v = heat of complete combustion of mixture at initial temperature T_0
$H_{v_{CO}}$ = heat of combustion of CO at initial temperature T_0
$H_{v_{H_2}}$ = heat of combustion of H_2 at initial temperature T_0
$\triangle u_{H_2}$ = change of thermal energy of H_2 between T_0 and T_e
$\triangle u_D$ = change of thermal energy of other diatomic gases between T_0 and T_e
e_0 = mols of O_2 remaining after combustion has proceeded to the equilibrium point
r = mols of N_2 in the mixture

* Eng. Exp. Sta. Bul. No. 139, p. 44.

† All heats of combustion are the lower heats of combustion in B.t.u. per mol.

This equation is of the form

$$y = b - ax \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

The water-gas equilibrium equation must also be satisfied, $H_2 + CO_2 \leftrightarrows H_2 + CO$. This equation expresses the relation which must exist between the CO, H_2, CO_2, and H_2O for any given temperature, and is

$$\frac{y\ (1-x)}{x\ (1-y)} = c \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

in which c is the equilibrium constant of the water-gas reaction, for the temperatures under consideration.

Eliminating y between equations (6) and (7)

$$a(c-1)\ x^2 + [a + b - c\ (b-1)]x - b = 0$$

a quadratic equation in x. Solving for x

$$x = \frac{-[a+b-c(b-1)] \pm \sqrt{[a+b-c\ (b-1)]^2 + 4ab\ (c-1)}}{2a\ (c-1)} \quad (8)$$

Referring to the tabular computation form (Table 1) the values of x corresponding to values of T of 4550, 4600, and 4700 deg. F. abs. are computed in Items 88a to 99. The values of c, the water-gas equilibrium constant, are obtained from Table 32, Bulletin No. 139.

From the energy equation (5) the value of y is determined for the values of x found (Item 100).

The equations of chemical equilibrium for both the CO reaction and the H_2 reaction must also be satisfied, but the use of either one of them, together with the two preceding equations, will completely determine the three variables T, x, and y.

Using the equilibrium equation for the CO reaction as given in Bulletin No. 139,

$$\log K_{p_{CO}} + \tfrac{1}{2} \log T = \log x + \tfrac{1}{2} \log \frac{M_1 T_0}{P_0} - \log\ (1-x)$$

$$- \tfrac{1}{2} \log\ (e_o - \frac{K}{2} x - \frac{L}{2} y) \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

in which

$K_{p_{CO}}$ = equilibrium constant for the reaction $CO + \tfrac{1}{2}O_2 \leftrightarrows CO_2$
M_1 = mols of gas in the charge
T_0 = temperature at the end of compression, deg. F. abs.
P_0 = pressure at end of compression, lb. per sq. ft. abs.

The left hand member of this equation is a function of T alone, or $L(T)$. The right hand member is a function of x and y alone, or $R(x,y)$.

Now, taking values of x and y as determined by equations (6) and (8), and substituting these in equation (9), values of $R(x,y)$ are found (Item 107). Values of $L(T)$ are also calculated, using the values of $K_{p_{CO}}$ from Table 30 in Bulletin No. 139, for the assumed temperatures 4550, 4600, and 4650 deg. F. abs. (Item 108).

The values of $R(x,y)$ and $L(T)$ are plotted against T in Fig. 2. The intersection gives values which satisfy all three equations (6), (8), and (9).

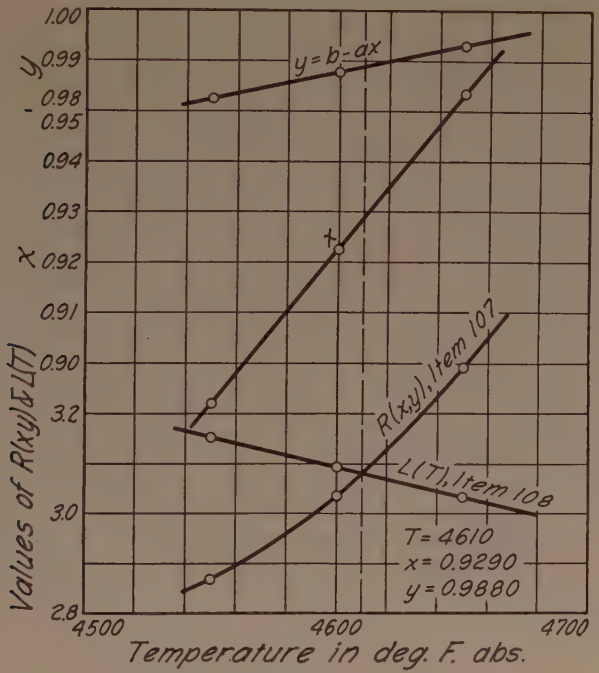

FIG. 2. VALUES OF T, $R(x,y)$, $L(T)$, x AND y FOR THE DETERMINATION OF MAXIMUM TEMPERATURE OF INCOMPLETE COMBUSTION

Plotting also the values of x and y as determined for the various temperatures and using the temperature determined by the intersection of the $R(x,y)$ and $L(T)$ curves, the following results are obtained for the conditions existing at the time of maximum temperature (Items 101, 102, and 103):

$$x = 0.9288$$
$$y = 0.9888$$
$$T''_e = 4610 \text{ deg. F. abs.}$$

That is, at the maximum temperature of the explosion, 4610 deg. F. abs., 92.88 per cent of the CO is burned to CO_2, and 98.88 per cent of the H_2 is burned to H_2O.

Calculating the pressure at the equilibrium point

$$P''_e = P_0 \frac{M''}{M_1} \frac{T''_e}{T_0} = 75.91 \times \frac{7.4916}{7.783} \times \frac{4610}{1400}$$

$$= 240.6 \text{ lb. per sq. in. abs. (Item 105)},$$

where M'' = mols of gas at equilibrium (Item 104)

M_1 = mols of gas in the initial charge (Item 20)

16. *Calculation of Adiabatic Expansion Curve.*—The calculation of the final temperature T_4 at the end of adiabatic expansion from maximum temperature calculated in Section 15 is based on the assumption that combustion is complete at the end of expansion, i. e., $x = y = 1.0000$ at T_4. For an adiabatic expansion the change of entropy is zero.

The method of calculation consists, therefore, of computing the entropy of the mixture at the equilibrium temperature $T_e = 4610$ deg. F. abs., and comparing this value with the entropy of the mixture (assuming $x = y = 1.0000$) at the expanded volume and at several assumed final temperatures. As the entropy remains constant, the final temperature can be determined by interpolation.

The assumption that $x = y = 1.0000$ at the end of expansion is justified by separate calculations which have been made, giving the values of T, x, and y at various points down the expansion curve. In all these calculations combustion was found to be complete at 50 per cent of the expansion stroke.

From the first law of thermodynamics

$$dq = du + Apdv \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

as before, where the work pdv is expressed in ft. lb., and A is the reciprocal of the conversion factor $J = 778$ ft. lb. per B.t.u. and also

$$du = \gamma_v dT \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

where γ_v is the instantaneous specific heat at constant volume of one mol of gas. This specific heat varies with the temperature according to the equation

$$\gamma_v = a + bT + fT^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

From the perfect gas law, for one mol of gas

$$Apv = RT$$

Differentiating

$$Apdv + Avdp = RdT \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

or

$$Apdv = RdT - Avdp$$

$$= RdT - RT\frac{dp}{p} \quad . \quad . \quad . \quad . \quad . \quad (14)$$

Substituting (11), (12), and (14) in (10), and dividing by T

$$\frac{dq}{T} = dS = \left[\frac{a+R}{T} + b + fT\right]dT - R\frac{dp}{p} \quad (15)$$

Integrating

$$S = S_0 + (a + R)\log_e T + bT + \frac{f}{2}T^2 - R\log_e p \quad (16)$$

where S_0 is the constant of integration.

As in Section 10, let

$$a\log_e T + bT + \frac{f}{2}T^2 = \phi$$

Then the entropy of one mol of any gas is

$$S = S_0 + \phi + R\log_e T - R\log_e p \quad . \quad . \quad . \quad (17)$$

The analysis of the gas mixture in the engine cylinder in the equilibrium state is

Constituent	Mols
CO_2	Kx
CO	$K(1-x)$
H_2O	Ly
H_2	$L(1-y)$
O_2	$(e_0 - \frac{K}{2}x - \frac{L}{2}y) = e''$
N_2	r
Total	$M'' = (K + L + e_0 + r) - \frac{K}{2}x - \frac{L}{2}y$

Multiplying equation (17) by the number of mols of each constituent existing at the equilibrium state gives the entropy of each constituent at the equilibrium state; adding these entropies, the entropy of the equilibrium mixture is

$$
\begin{aligned}
S = {} & Kx\,(S_{0CO_2} + \phi_{CO_2} + R\log_e T) - Kx\,R\log_e \frac{KxP}{M''} \\
& + K\,(1-x)\,[S_{0_{CO}} + \phi_{CO} + R\log_e T] \\
& \quad - K\,(1-x)\,R\log_e \frac{K\,(1-x)\,P}{M''} \\
& + Ly\,(S_{0_{H_2O}} + \phi_{H_2O} + R\log_e T) - Ly\,R\log_e \frac{LyP}{M''} \\
& + L(1-y)\,[S_{0_{H_2}} + \phi_{H_2} + R\log_e T] \\
& \quad - L(1-y)\,R\log_e \frac{L\,(1-y)\,P}{M''} \\
& + e''\,(S_{0_{O_2}} + \phi_{O_2} + R\log_e T) - e''\,R\log_e \frac{e''\,P}{M''} \\
& + r\,(S_{0_{N_2}} + \phi_{N_2} + R\log_e T) - rR\log_e \frac{rP}{M''}
\end{aligned}
\qquad (18)
$$

where P is the total pressure. The pressure used for each constituent is, of course, the partial pressure of that constituent, expressed in terms of the total pressure.

Rearranging,

$$
\begin{aligned}
S = {} & -Kx\,[(S_{0_{CO}} + \tfrac{1}{2}S_{0_{O_2}} - S_{0_{CO_2}}) + (\phi_{CO} + \tfrac{1}{2}\phi_{O_2} - \phi_{CO_2})] \\
& - Ly\,[(S_{0_{H_2}} + \tfrac{1}{2}S_{0_{O_2}} - S_{0_{H_2O}}) + (\phi_{H_2} + \tfrac{1}{2}\phi_{O_2} - \phi_{H_2O})] \\
& + KS_{0_{CO}} + LS_{0_{H_2}} + e_0 S_{0_{O_2}} + rS_{0_{N_2}} + K\phi_{CO} + L\phi_H \\
& \quad + e_0\,\phi_{O_2} + r\,\phi_{N_2} \\
& + M''R\log_e T + M''R\log_e M'' - M''R\log_e P \\
& - R\,[Kx\log_e Kx + K\,(1-x)\log_e K\,(1-x) + Ly\log_e Ly \\
& + L(1-y)\log_e L(1-y) + e''\log_e e'' + r\log_e r]
\end{aligned}
\qquad (19)
$$

From the perfect gas law

$$P = \frac{1544\,M''T}{V}$$

in which 1544 is the universal gas constant $R = 1.985$, expressed in ft. lb. per degree F.

Then, taking natural logarithms and multiplying by $M''R$,

$$M''R\log_e P = M''R\log_e M'' + M''R\log_e T + M''R\,(\log_e 1544 - \log_e V) \qquad (20)$$

Let

$$\left.\begin{aligned}\alpha &= (S_{0_{CO}} + \tfrac{1}{2} S_{0_{O_2}} - S_{0_{CO_2}}) + (\phi_{CO} + \tfrac{1}{2}\phi_{O_2} - \phi_{CO_2}) \\ \beta &= (S_{0_{H_2}} + \tfrac{1}{2} S_{0_{O_2}} - S_{0_{H_2O}}) + (\phi_{H_2} + \tfrac{1}{2}\phi_{O_2} - \phi_{H_2O})\end{aligned}\right\} \quad (21)$$

Since the mean specific heats of CO, O_2, and N_2 have the same numerical value,* the expressions for these three gases are grouped together with the subscript D (diatomic). With the substitution of (20) and (21) in equation (19), the expression for the entropy becomes

$$\left.\begin{aligned}S = &- Kx\alpha - Ly\beta + (K + e_0 + r)\phi_D + L\phi_{H_2} \\ &- 4.571\, M'' (\log 1544 - \log V) \\ &- 4.571\, [Kx \log Kx + K(1 - x) \log K(1 - x) \\ &\quad + Ly \log Ly + L(1 - y) \log L(1 - y) \\ &\quad + (e_0 - \frac{K}{2}x - \frac{L}{2}y) \log (e_0 - \frac{K}{2}x - \frac{L}{2}y)] \\ &+ [KS_{0_{CO}} + LS_{0_{H_2}} + e_0 S_{0_{O_2}} + rS_{0_{N_2}} = 4.571\, r \log r]\end{aligned}\right\} \quad (22)$$

in which logarithms are taken with the base 10.† The units of pressure, temperature, and volume are:

P = pressure, lb. per sq. ft. abs.
T = temperature, deg. F. abs.
V = volume in cubic feet

The last term of equation (22) is constant for all temperatures, and is therefore neglected since the change of entropy alone is considered.

The value of S, the entropy of the gaseous mixture at equilibrium, at the maximum temperature of 4610 deg. F. abs. is calculated in Items 109 to 115. The value of S is found to be 327.9488. After taking $V = V_4$, the volume at the end of the adiabatic expansion, values of T_4 are assumed, namely, 3100, 3200, and 3300 deg. F. abs. The entropy of the mixture is calculated for each of these temperatures, assuming $x = y = 1.0000$, i.e., that combustion is complete. By interpolation between the values of S for 3200 and 3300 degrees, using the value computed, $S = 327.9488$, the temperature T_4 is found to be 3209 deg. F. abs. (Item 118).

From the perfect gas law

$$P_4 = 33.34 \text{ lb. per sq. in. abs.} \quad \text{(Item 120)}$$

---

* Eng. Exp. Sta. Bul. No. 139.

† The factor 4,571 is the product of the conversion factor of $\log_e x$ to $\log_{10} x$, or 2.30259, and R, which is 1.985.

17. *Efficiency of Ideal Cycle.*—The work done during the whole adiabatic cycle is given by

$$ {}_1W_4 = U_{m_1} - U_{p_4} $$

where

${}_1W_4$ = the work done during the ideal adiabatic Otto cycle
U_{p_4} = the total energy of the products of combustion at the end of the expansion stroke
U_{m_1} = the total energy of the gases at the beginning of the compression stroke

Let U_{p_1} = the total energy of the products of combustion at the initial temperature T_1 ($= T_{100}$).

Adding and subtracting U_{p_1}

$$ {}_1W_4 = U_{m_1} - U_{p_1} - (U_{p_4} - U_{p_1}) $$

Substituting $H_{v_{T_1}}$ for $(U_{m_1} - U_{p_1})$ the expression for work becomes

$$ {}_1W_4 = H_{v_{T_1}} - (U_{p_4} - U_{p_1}) \quad . \quad . \quad . \quad . \quad (23) $$

For incomplete combustion this expression becomes

$$ {}_1W_4 = H_{v_{T_1}} - (1 - x_4) H_{v_{CO_{T_1}}} - (1 - y_4) H_{v_{H_{2_{T_1}}}} - (U_{p_4} - U_{p_1}) \quad (24) $$

where

$$ U_{p_4} = Kxu_{CO_2} + Lyu_{H_2O} + L(1 - y)u_{H_2} $$

$$ + (K + e_0 + r - \frac{3}{2}Kx - \frac{3}{2}Ly)u_D \quad . \quad . \quad . \quad . \quad (25) $$

In these equations

x_4 = degree of completion of CO reaction at end of expansion.
y_4 = degree of completion of H_2 reaction at end of expansion.
$H_{v_{T_1}}$ = heat of combustion of the mixture at temperature T_1 at the beginning of compression.

The basis of computing efficiency is taken as the heat of complete combustion of the fuel at 520 deg. F. abs. Therefore, the efficiency of the adiabatic cycle is

$$ \eta = \frac{{}_1W_4}{H_{v_{520}}} = \frac{H_{v_{T_1}} - (1 - x_4) H_{v_{CO_{T_1}}} - (1 - y_4) H_{v_{H_2}} - (U_{p_4} - U_{p_1})}{H_{v_{520}}} \quad (26) $$

For those cases in which complete combustion is attained at the end of the expansion this expression reduces to

$$\eta = \frac{H_{vT_1} - (U_{p_4} - U_{p_1})}{H_{v_{520}}} \quad . \quad . \quad . \quad . \quad . \quad (27)$$

Since H_{vT_1}, U_{p_4}, and U_{p_1} are all functions of temperature alone, it is evident that the efficiency is directly dependent on the initial and final temperatures.

18. *Calculation of Ideal Efficiency for Test No. 22.*—The temperature at the beginning of the compression stroke for the ideal cycle is 779 deg. F. abs. (Section 12 and Item 71). The heat of combustion at 779 deg. F. abs. is 182 247 B.t.u. per mol (Item 40). The thermal energy of the products of combustion is calculated as before, at the two temperatures $T_1 = 779$ deg. F. abs. and $T_4 = 3209$ deg. F. abs. At T_1, $U_p = 31\,200$ B.t.u. per mol, and at T_4, $U_p = 146\,426$ B.t.u. per mol (Item 83b).

The heat of combustion at 520 deg. F. abs. is 182 343 B.t.u. per mol (Item 3).

$$_1W_4 = 182\,247 - (146\,426 - 31\,200) = 67\,021 \text{ B.t.u.}$$

$$\eta = \frac{67\,021}{182\,343} \times 100 = 36.76 \text{ per cent (Item 87b)}$$

19. *Efficiency of Ideal Cycle Assuming Complete Combustion.*—The ideal thermal efficiency can be computed assuming that the fuel is completely burned at constant volume, i.e., $x = y = 1$ at the end of the constant volume explosion. The temperature attained in the case of complete combustion was 4722 deg. F. abs. (Item 75) and the corresponding temperature at the end of adiabatic expansion was 3199 deg. F. abs. (Item 81).

Computing the ideal thermal efficiency according to Section 18 (Item 83b), the work of the cycle is

$$182\,247 - (145\,874 - 31\,200) = 67\,573 \text{ B.t.u.}$$

The ideal thermal efficiency is then

$$\eta' = \frac{67\,573}{182\,343} \times 100 = 37.06 \text{ per cent (Item 86).}$$

20. *Effect of Dissociation on Efficiency.*—The calculated ideal efficiencies for Test No. 22 are

η (with dissociation)	= 36.76 per cent
η' (without dissociation)	= 37.06 per cent
Difference . . .	= 0.30 per cent

The values of x and y at maximum temperature are

$$x = 0.9288$$
$$y = 0.9888$$

In this particular case the effect of dissociation is practically negligible because of the high percentage of excess air and neutral exhaust gas in the charge.

The ideal indicator diagram as calculated and the actual diagram are shown in Fig. 3.

III. Thermodynamics of Actual Cycle

21. *Indicated Thermal Efficiency.*—The losses due to restriction of gas flow by the valves are eliminated by calculating the thermal efficiency, using the area under the combustion and expansion line minus the area under the compression line as representing the work done. The effect of the early opening of the exhaust valve is eliminated by extending the expansion line to the end of the stroke. Also, the calculated adiabatic compression curve is used in the actual case for the range 70 to 100 per cent of the stroke in order to eliminate the effect of the low suction pressure. The calculated curve practically coincides with the actual curve in this range.

The scales used in plotting the original indicator diagrams from the test data are:

Ordinates, 1 inch = 20 lb. per sq. in.
Abscissas, 1 inch = 10 per cent of piston stroke.

The volume of the charge of 7.783 mols in the cylinder at the end of compression is

$$V_0 = \frac{MRT_0}{P_0} = \frac{1544 \times 7.783 \times 862}{75.91 \times 144} = 1539.1 \text{ cu. ft.}$$

The work represented by one square inch of indicator diagram area is then

$$1 \text{ sq. in.} = \frac{20 \times 144 \times 0.10 \times 4 \times 1539.1}{778} = 2279.5 \text{ B.t.u.}$$

As the compression ratio in Test No. 22 is 5.00, the volume increase for 1 inch as abscissa on the indicator diagram is $0.10 \times 4 \times 1539.1$ cu. ft.

The area between the expansion and compression line, as described above, is 21.92 sq. in., for Test No. 22. The indicated

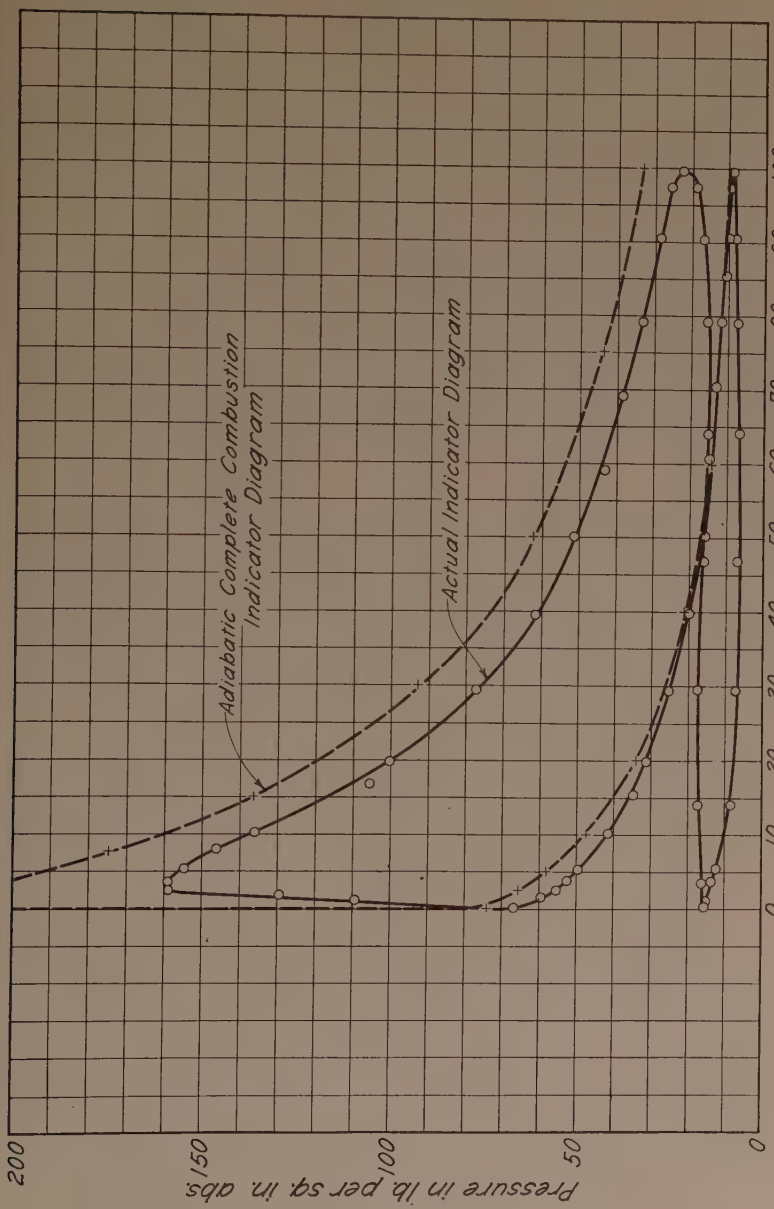

FIG. 3. ACTUAL AND IDEAL INDICATOR DIAGRAMS FOR TEST NO. 22

thermal efficiency of compression, explosion, and expansion is then

$$\eta_c = \frac{21.92 \times 2279.5}{182\,343} \times 100 = 27.40 \text{ per cent}$$

The indicated thermal efficiency as computed from the total net area of the indicator card is

$$\eta_i = \frac{17.533 \times 2279.5}{182\,343} \times 100 = 21.92 \text{ per cent}$$

There is a loss of 5.48 per cent, therefore, due to throttling in the valves and to running at partial load.

22. *Factors Affecting Indicated Thermal Efficiency.*—Eliminating the effect of valve restriction, the actual efficiency is only 27.40 per cent while the ideal efficiency for the same case is 36.76 per cent. The difference, 9.36 per cent, results from two causes, (1) slow burning and (2) heat loss.

23. *Method of Determining Time of Reaction and Heat Loss.*—The heat loss as measured by the increase in temperature of the jacket water represents the total losses, minus the radiation from the jacket, for all phases of the cycle. A large amount of heat is transmitted to the jacket water during the exhaust stroke, and afterwards to the water-jacketed portions of the exhaust pipe. The inclusion of these indeterminate quantities of heat in the heat loss as shown by the rise in temperature of the jacket water render it difficult to draw any valid conclusions from heat losses calculated from jacket water.

From data obtained from the indicator diagram, the weight of charge, and the initial temperature of the charge within the cylinder, it is possible to compute the approximate heat loss during the various phases of the cycle, and also to estimate the time required for combustion.

24. *Heat Loss during Compression.*—As shown in Fig. 3 the compression process from 100 per cent (the start of compression) up to 70 per cent piston stroke may be considered as adiabatic without appreciable error. Since ignition occurs in some of the tests as early as 15 per cent of the stroke from dead center, heat losses past this point cannot be calculated from the actual compression curve. The heat loss during compression is therefore defined as the heat loss during the compression from 70 per cent to 15 per cent of the stroke.

From the first law of thermodynamics the heat loss during compression, with no chemical reaction occurring, is

Heat loss = work done — increase in thermal energy.

In a given case the work done is obtained by determining the area under the compression curve for the range 70 per cent to 15 per cent by means of a planimeter. The increase in thermal energy is obtained by making use of the temperatures calculated by applying the perfect gas law to the pressure, volume, and weight of charge existing for the given state.

For Test No. 22:

Area under compression curve, 70 per cent to 15 per cent stroke = 5.555 sq. in. (Item 45)

Work of compression = $5.555 \times 2279.5 = 12\,663$ B.t.u. (Item 46)

Temperature at 70 per cent stroke = 862 deg. F. abs. (Item 24)

Pressure at 15 per cent stroke = 34.7 lb. per sq. in. (Item 26)

Temperature at 15 per cent stroke =

$$T_{15} = T_{70} \times \frac{P_{15}}{P_{70}} \times \frac{V_{15}}{V_{70}} = 862 \times \frac{34.7}{12.3} \times \frac{1+0.15\times4}{1+0.70\times4} = 1024 \text{ deg. F. abs.}$$ (Item 27)

The increase of thermal energy of the charge between 862 and 1024 deg. F. abs. is computed in Items 42, 43, and 47, and is found to be 6760 B.t.u. The heat loss between 70 per cent and 15 per cent compression stroke is then

$$12\,663 - 6760 = 5903 \text{ B.t.u.} \quad \text{(Item 48)}$$

or, expressing the loss as a percentage of the heat of combustion of the charge at 520 deg. F. abs., it is

$$\frac{5903}{182\,343} \times 100 = 3.24 \text{ per cent} \quad \text{(Item 48)}$$

The heat loss during compression as computed above is added to the loss during explosion and expansion as determined later.

25. *Heat Loss and Reaction Velocity during Combustion and Expansion.*—In the actual combustion in the engine the velocity of the reaction is slow during the period immediately following ignition. As the flame spreads the velocity of reaction increases, and rapid combustion proceeds until the reaction is nearly complete. The remaining parts of the charge burn with a fairly low velocity.

Assuming that it is possible to calculate the extent of combustion at any given point on the cycle, the heat loss from the time of ignition can be calculated from the equation

Heat loss = heat of partial combustion — work done —increase of thermal energy; or

$$H = zH_v - W - \Delta U_p \quad . \quad . \quad . \quad . \quad . \quad . \quad (28)$$

At present there is not sufficient information available to permit the exact calculation of the extent of combustion at a given time after ignition; however, an approximate estimate can be made at a given point in the cycle.

Let

ΔU_p = increase of thermal energy of products of combustion

H_v = total heat of combustion at the initial state (15 per cent on compression stroke)

H = total heat loss from time of ignition to any given point in the cycle

z = fraction of H_r evolved by the reaction up to the given point in the cycle.

Adding $(1 - z)\ H_r$ to each side of the preceding equation

$$H + (1 - z)H_v = H_v - W - \Delta U_p \quad . \quad . \quad . \quad . \quad (29)$$

The work done from the point of ignition to any given point in the cycle can be determined from the indicator diagram. By assuming complete combustion the temperature at any point on the indicator diagram can be calculated from the pressure, volume, and weight of charge. For the points on the early part of the combustion curve the calculated temperature may be as much as 4.5 per cent too high, because of the assumed reduction in the number of mols which has not yet occurred. For the points near the end of the combustion curve the error in the calculated temperature is negligible.

For Test No. 22 the heat loss from 15 per cent on compression to 30 per cent on the expansion curve plus the unused part of the heat of combustion is calculated as follows:

The work area of the form shown in Fig. 4 is 13.781 sq. in., and represents 31 414 B.t.u (see p. 34). From the indicator diagram, the pressure at 30 per cent on the expansion stroke is 76.0 lb. per sq. in. abs. The volume is

Clearance + 30 per cent piston displacement
= 0.1017 + 0.30 × 0.4068 = 0.2237 cu. ft.

Assuming complete combustion, the number of mols of gas is 7.457 (Item 37). The actual number of mols of gas present before

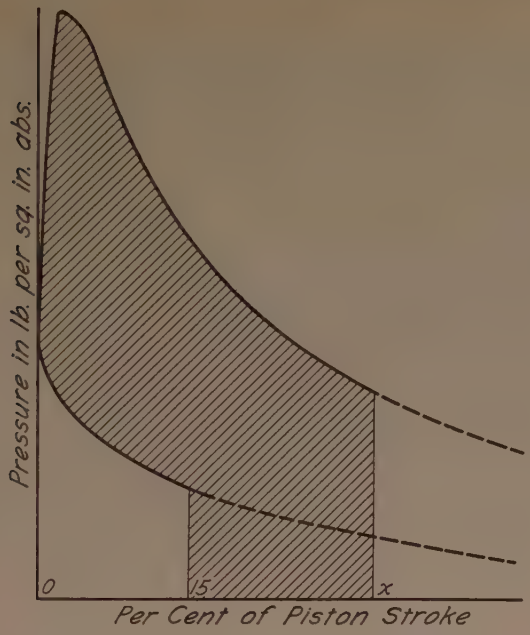

FIG. 4. AREA ON INDICATOR DIAGRAM FROM 15 PER CENT ON COMPRESSION STROKE TO x PER CENT ON EXPANSION STROKE

ignition was 0.00051416 (Item 16). Then the temperature is

$$T_{30} = \frac{PV}{MR} = \frac{76.0 \times 144 \times 0.2237 \times 7.783}{0.000\,514\,16 \times 1544 \times 7.457}$$

$$= 3219 \text{ deg. F. abs.}$$

The increase in thermal energy is calculated as before in Items 58 and 59, and is 146 978 — 41 450 = 105 528 B.t.u. Taking the heat of combustion for conditions at 15 per cent compression stroke, H_r = 182 175 (Item 38a) and subtracting the change of energy and the work done

$$\begin{aligned} H + (1 - z)\, H_r &= H_v - W - \Delta U_p \\ &= 182\,175 - 31\,414 - 105\,528 \\ &= 45\,233 \text{ B.t.u.} \end{aligned}$$

Or, expressing $[H + (1 - z)H_v]$ in per cent of heat of combustion at 520 deg. F. abs., it is

$$\frac{45\,233}{182\,343} \times 100 = 24.76 \text{ per cent}$$

Similarly, values of $[H + (1 - z) H_v]$ are calculated for other points on the expansion curve. (See Items 48 and 51 to 62.)

When the calculated values of $[H + (1 - z) H_v]$ are plotted against the per cent of stroke, an analysis of the curves gives information regarding the extent of combustion and the rate of heat loss during the explosion and expansion processes.

IV. Description of Testing Plant

26. *General Arrangement of Testing Plant.*—The gas engine and testing apparatus, shown in Fig. 5, consisted of

(a) the engine
(b) the dynamometer
(c) the gas storage and measuring tanks
(d) the air measuring orifice and other instruments

A view of the engine cylinder with the indicator *A*, the indicator timer *B*, and the ignition timer *C*, is shown in Fig. 6. A diagram of piping connections is shown in Fig. 7.

27. *Description of Engine.*—The engine used for the tests was a 9 in. x 10 in. single acting, single cylinder horizontal engine. It was designed to run on gas or gasoline, but for the tests described in the bulletin the gasoline attachments were removed. The principal dimensions of the engine are as follows:

Bore	10 in.
Stroke	9 in.
Piston displacement	0.4068 cu. ft.
Length of piston	11½ in.
Valve lift	½ in.
Length of connecting rod	20 in.
Diameter of piston rod	1½ in.
Mean diameter of valve seats	3 ¾ in.
Lubrication	drip cups

Cooling—circulating water through jacket and cylinder head.

Sectional views of the engine cylinder are shown in Fig. 8.

The valves were of the poppet type, and were operated by a single cam on a lay shaft geared to run at half engine speed.

The clearance volume was changed by screwing the piston rod into the crosshead. A determination of the actual clearance volume was made at one compression ratio by filling the cylinder with a weighed amount of oil of known specific gravity. The clearance volume for any given setting of the rod could be calculated from the initial volume, the pitch of the threads, and number of turns of the rod. These calculated compression ratios were also checked by actual measurement with oil.

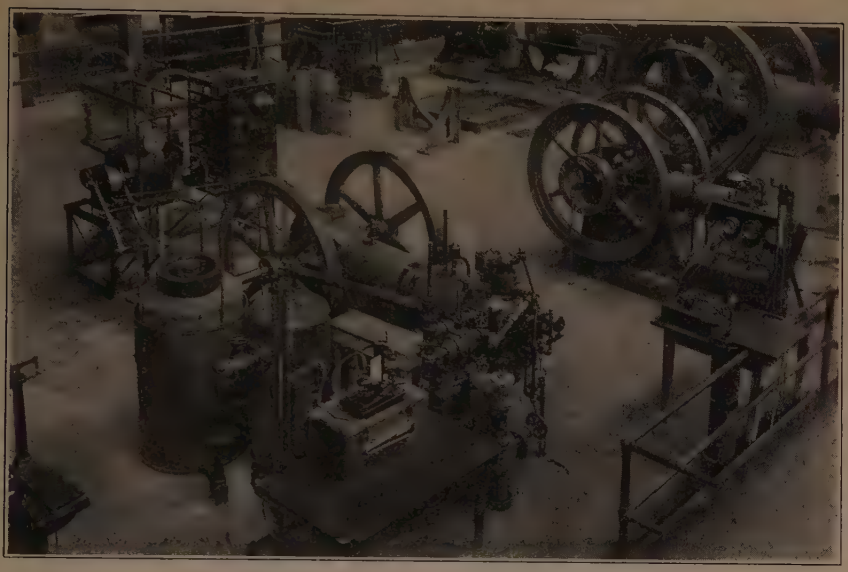

FIG. 5. ENGINE AND AUXILIARY APPARATUS

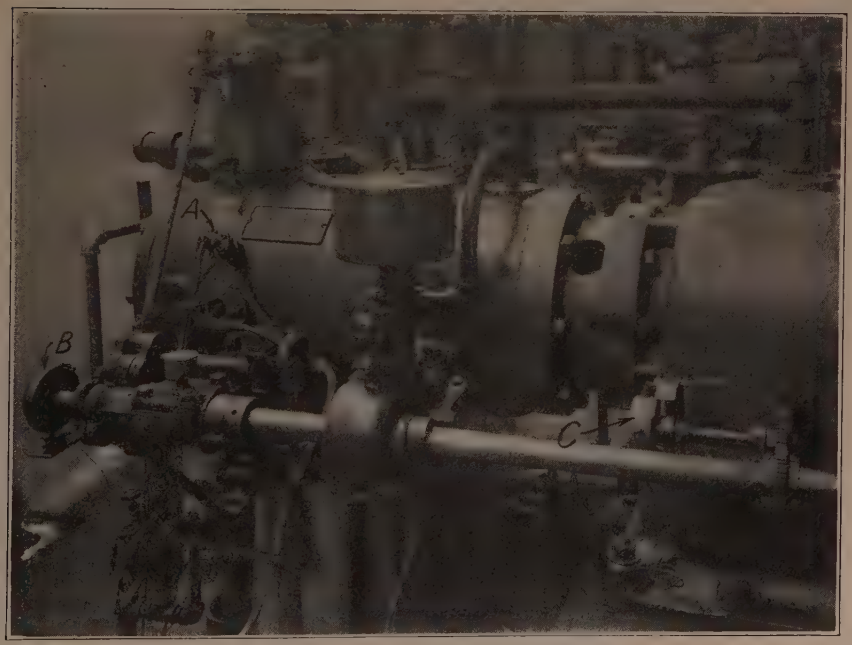

FIG. 6. ENGINE CYLINDER AND INDICATOR

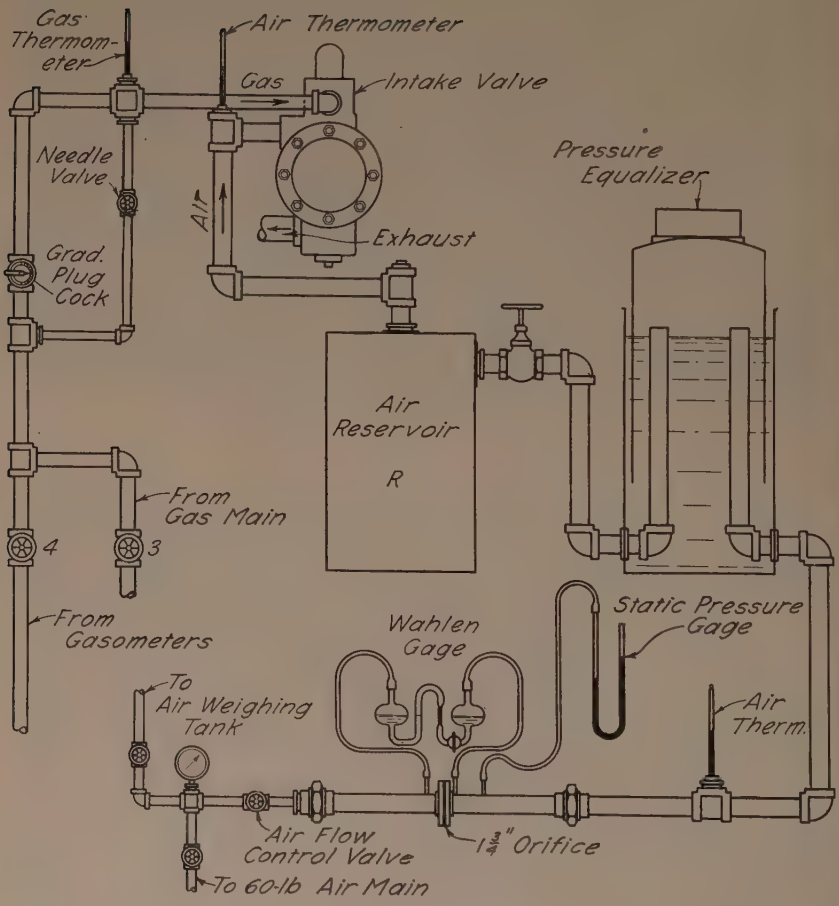

FIG. 7. DIAGRAM OF PIPING CONNECTIONS

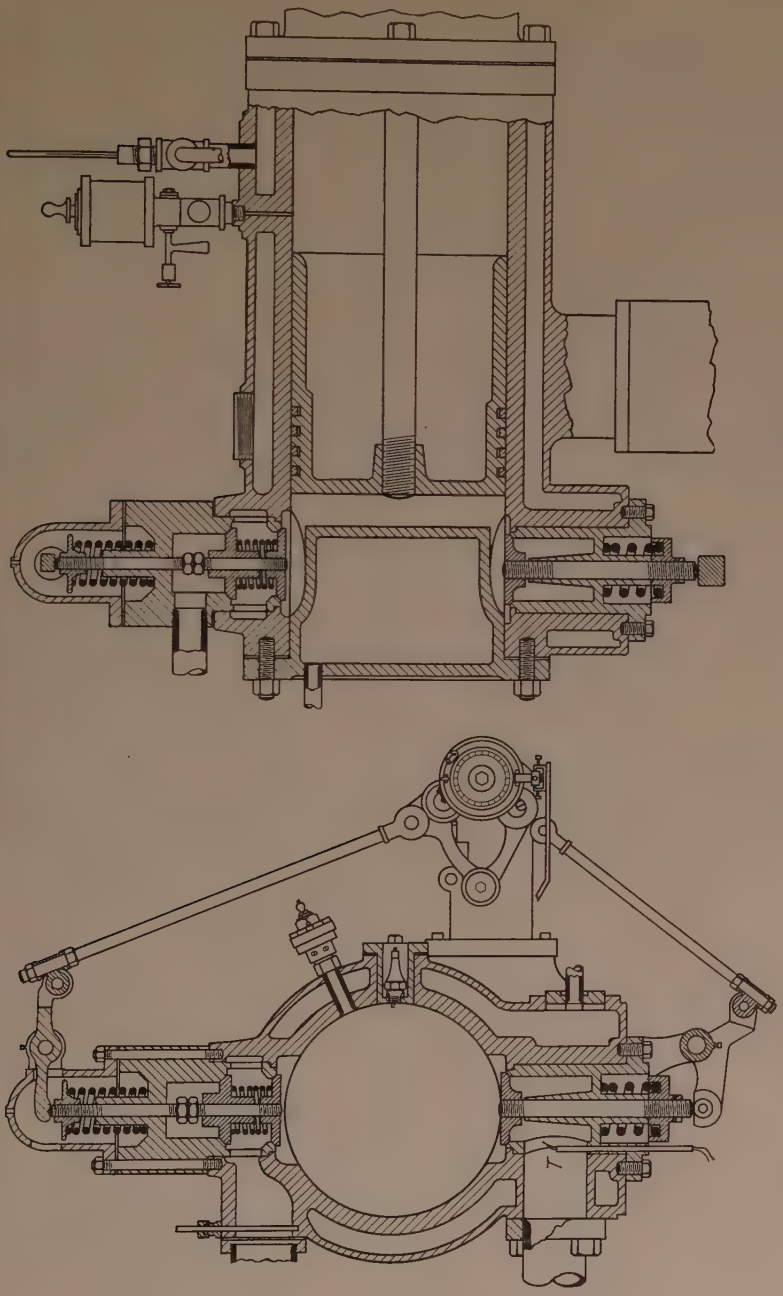

FIG. 8. SECTIONAL VIEWS OF ENGINE CYLINDER

28. *Gas and Air Valve.*—A considerable amount of trouble was experienced during the early tests on account of backfiring through the intake valve, particularly when running with lean mixtures. To eliminate this trouble a special gas and air intake valve was constructed and installed in place of the original intake valve. The details of this valve are shown in Fig. 8. It operated so that the air valve opened earlier and closed later than the gas valve. The stream of air admitted before and after the admission of the gas swept the passages out and obviated any chance of backfiring. No trouble from backfiring or pre-ignition was experienced after the installation of this valve.

29. *Ignition.*—After some difficulty with various ignition systems which would not function consistently over the wide range of fuel mixtures employed, a high tension spark system was finally adopted. A standard $\frac{1}{2}$-in. spark plug, screwed into the cylinder as shown in Fig. 8, was connected with the high tension coil. The timer was driven from the half-speed shaft, and could be adjusted to vary the time of ignition from 65 degrees before to 15 degrees after dead center.

30. *Dynamometer.*—The engine was belted to a 10-25 horsepower electric absorption dynamometer. After warming up, it was loaded to approximately 10 horsepower by means of the dynamometer. This load was maintained as nearly constant as possible during the test. Since the engine speed did not vary more than 2 per cent above or below the desired speed during any one test, it is evident that the load remained practically constant.

No attempt was made to determine the efficiency of the driving belt or of the dynamometer, since the computations were based on the indicator diagram rather than the power output of the engine.

31. *Gas Measurement.*—The city gas used as a fuel for the tests was taken directly from the mains and stored in two large gasometers having a capacity of approximately 160 cu. ft. each. They were provided with scales graduated accurately to inches, so that the position of the gasometer bell and consequently the volume of gas could be determined at any time. Since the water in the gasometers was at room temperature as were also the gasometers themselves, this method insured a supply of gas at constant temperature.

32. *Method of Determining Gas Consumption.*—Previous to running a test the gasometers were filled from the mains, and then cut off from the gas supply. The engine was warmed up by running on gas from the mains. Valve 3 (Fig. 7) was then closed, and valve 4 from gasometer 1 was opened simultaneously, the engine then drawing its fuel from gasometer 1. After obtaining the desired

test conditions the fuel supply was changed from gasometer 1 to gasometer 2, and the test was started.

The time at which the scale index on gasometer 2 passed a given scale graduation was noted. After taking the indicator card and all other required data for one complete test, the time of passing another convenient scale graduation was noted. The pressure and temperature of the gas in the gasometer were read at intervals during the test. These data gave the volume of gas consumed by the engine in a given length of time. In most cases a total volume of about 140 cu. ft. of gas was used during the period over which data were taken.*

33. *Method of Calculation of Gas Consumption in Standard Cubic Feet.*—The cubic foot of standard gas is defined as the cubic foot at 60 deg. F. and pressure corresponding to 29.92 in. of mercury. The calibration constants of the gasometer and the corrections to be applied to reduce the consumption of gas to standard conditions, are as follows:

A consideration of the pressures and volumes of water and gas shows that the water level outside the gasometer bell remains constant for all positions of the bell. There is a slight elevation of the water level inside the bell due to the water displaced by the metal walls. This elevation amounted to 0.5 in. for a 60-inch drop of the bell. Consequently, a correction proportional to the actual drop of the bell was applied.

The pressures inside the bell also varied somewhat, due to the decreased effective weight of the bell when sinking into the water. A correction for this variation is applied to the total atmospheric pressure. The mean excess of pressure inside the bell over atmospheric pressure was 0.35 in. of mercury, and, of course, was constant for all tests.

We have then

$$V_1 - V_2 = A(H_1 - H_2 + \triangle h)$$

where

V_1 = initial volume in cu. ft.
V_2 = final volume in cu. ft.
A = net area of gasometer bell in sq. ft.
H_1 = initial height of bell in ft.
H_2 = final height of bell in ft.

$\triangle h$ = correction for change of water level inside bell $= 0.5 \times \frac{H_1 - H_2}{5}$

* All valves and piping in the gas line were tested for leaks. No leakage was found over a period of several hours.

The volume of gas used at 29.92 inches barometric pressure and 60 deg. F. is given by

$$V_s = (V_1 - V_2)_s = \frac{T_s}{T} \frac{P_1V_1 - P_2V_2}{P_s}$$

where the subscripts s denote volumes, pressures, and temperatures under the standard conditions of 29.92 in. barometric pressure and 60 deg. F. P_1 and P_2 are almost equal, and hence the mean of the two* is used. Then

$$V_s = \frac{T_s}{TP_s} \frac{P_1 - P_2}{2} (H_1 - H_2 - \triangle h) A$$

34. *Calculations for Gas Consumption on Test No. 22.*—We have the following data:

Drop of gasometer bell	58.00 in.
Time	30.57 min.
Barometric pressure	29.30 in.
Temperature (mean)	89.30 deg. F.
Mean of pressures	$29.30 + 0.35$ = 29.65 in.

$$V_s = \frac{459.6 + 60}{459.6 + 89.3} \frac{29.65}{29.92} (58 - 0.5)$$
$$= 129.2 \text{ cu. ft.}$$

The gas consumed per minute was then

$$\frac{129.2}{30.57} = 4.227 \text{ cu. ft.}$$

measured at 29.92 in. barometric pressure and 60 deg. F.

An examination of the precision of the various measurements made showed that the determination of the gas consumption is accurate to within one per cent.

35. *Air Measuring Orifice.*—Air was supplied to the engine intake from a main which carried air at 60 lb. pressure.

A thin plate orifice, inserted between pipe flanges (Fig. 7) was used to measure the air consumption. The orifice was made of 16-gage sheet steel. The flow of air to the engine was controlled by a valve. In order to damp out the oscillations of the air passing through the orifice a small gasometer was connected between the orifice and the engine intake, a throttle valve being placed in the line (Fig. 7). The gasometer bell was weighted to give the desired pressure at the intake valve, and was kept floating by the proper admission of

* A calculation of the error introduced by using the mean of the initial and final pressures showed that for a very extreme case the error amounted to less than 0.1 per cent.

compressed air from the main. By adjustment of the throttle valve and the weight on the gasometer bell, practically no fluctuation of pressure was noticeable at the orifice.

A Wahlen gage (Illinois micromanometer)* was used to measure the pressure differential across the orifice. A mercury manometer connected as in Fig. 7 showed the static pressure in the pipe line at the orifice. The air temperature was read by means of a thermometer.

36. *Calibration of Orifice.*—The orifice was calibrated in place by air weighed from the "Air Weighing Plant."† The calibration process consisted of weighing 150 lb. of air drawn from a large tank on scales, and delivering it to the orifice through a pipe line.‡ A calibration curve for the orifice is given in Fig. 9.

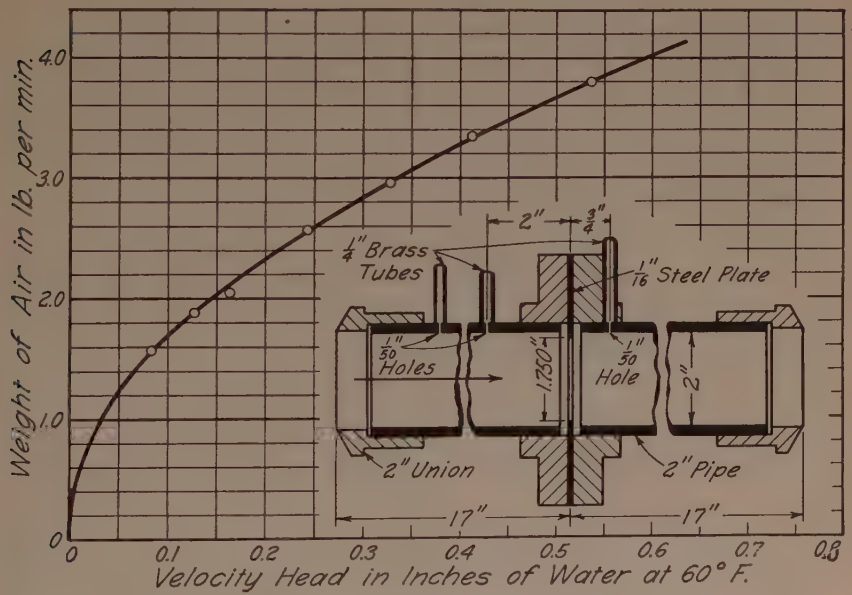

Fig. 9. Orifice Calibration Curve and Details of Orifice

37. *Use of Orifice.*—At least five readings of pressure differential at the orifice were taken during each test. At the same time the static

* "Investigation of Warm Air Furnaces and Heating Systems," Univ. of Ill. Eng. Exp. Sta. Bul. 120, 1921, pp. 88-103.

† "Investigation of Warm Air Furnaces and Heating Systems," Univ. of Ill. Eng. Exp. Sta. Bul. 120, pp. 81-88.

‡ All the air piping for calibration purposes was carefully tested and found to be free from leaks.

pressure and temperature of the air were observed. Since the engine was operated under constant conditions, the rate of flow of air through the orifice and consequently the pressure differential also remained constant, seldom varying more than 0.3 per cent from the mean of the five readings. The relative humidity of the air supplied to the engine from the main was determined by means of an Assman aspirating psychrometer immediately after the close of each test.

38. *Calculation of Air Consumption in Standard Cubic Feet.*— The air consumption was based on the cubic foot of air at 60 deg. F. and pressure corresponding to 29.92 in. of mercury.

The following data were taken for the determination of the air consumption in Test No. 22:

Differential pressure at orifice	0.105 in. water
Air temperature (mean)	96.50 deg. F.
Barometric pressure	29.30 in. Hg.
Static pressure	1.48 in. Hg.
Humidity	33 per cent at 89 deg. F.

From the calibration curve for the orifice (Fig. 9) it is found that a pressure differential of 0.105 inches is equivalent to a flow of 1.730 lb. of air per minute, at 70 deg. F. and 29.92 in. pressure. The actual weight of air flowing then is

$$1.730 \times \sqrt{\frac{29.30 + 1.48}{29.92} \times \frac{459.6 + 70.0}{459.6 + 96.5}} = 1.712 \text{ lb. per minute}$$

This computation gives the flow of moist air (33 per cent relative humidity at 89 deg. F.) in pounds per minute.

The weight of saturated water vapor per pound of dry air at 89 deg. F. is 0.03008 lb.* The actual weight of water vapor per pound of dry air is then $0.33 \times 0.3008 = 0.00993$ lb. The weight of water vapor required to saturate 1 pound of dry air at 60 deg. F. (the standard temperature) is 0.01105 lb.† The volume of 1 pound of dry air plus the water to saturate it at 60 deg. F. is 13.33 cu. ft., while the volume of 1 pound of dry air alone at 60 deg. F. is 13.10 cu. ft. The volume occupied by the water vapor is then 13.33 — 13.10 or 0.23 cu. ft. for the saturated condition. The actual volume of water vapor is then

$$\frac{0.00993}{0.01105} \times 0.23 = 0.21 \text{ cu. ft.}$$

The volume of the 1.011 lb. of mixture of air and water vapor is then $13.10 + 0.21 = 13.31$ cu. ft. at 60 deg. F. and 29.92 in. pressure.

* G. A. Goodenough, "Properties of Steam and Ammonia," 1917, p. 85.
† Ibid.

The standard flow in cubic feet of mixture (air and water vapor) per minute is then

$$\frac{13.31 \times 1.712}{1.011} = 22.561 \text{ cu. ft. per min.}$$

at 60 deg. F. and 29.92 in. barometric pressure.

The air-gas ratio for the test is computed from this value and the gas consumption (Section 34) and is expressed in volumes of air per volume of gas, both measured at 60 deg. F. and 29.92 in. barometric pressure. For Test No. 22 the ratio is

$$\frac{22.561}{4.227} = 5.337$$

39. *Description of Indicator.*—The indicator used for the tests was of the balanced diaphragm type.* Inasmuch as the accuracy of the results obtained from the tests depended largely upon the precision of the indicator and the accuracy of the indicator diagrams, the balanced diaphragm indicator was selected for use even though the engine speed was not high. The indicator was connected directly to the engine cylinder without any indicator cock intervening.

Briefly, the operation of the indicator consists in balancing the pressure in the cylinder at any desired instant of the cycle by a known pressure applied from an outside high pressure source. The equilibrium condition is determined by the cessation of "clicks" in a telephone receiver connected in series with the diaphragm and a contact placed very close to it. A coördinating device, or timer, is then adjusted to a new position in the cycle, and the balancing operation is repeated. Thus a number of separate observations of the pressures existing in the engine cylinder at different times in the cycle are made, and a diagram is plotted from the data thus obtained. It is evident that such a graph is an average diagram, showing the average area over a considerable period of time. Therefore the one diagram, the data for which were secured in about 30 minutes, was used as the average for the whole test. As measurements of the other quantities involved were made only during the time of taking the diagram and as the operating conditions of the engine were kept practically constant during this time, a diagram taken in the manner described represents actual average conditions with considerable accuracy.

40. *Precision and Accuracy of Indicator.*—The balancing pressures were read by means of four gages covering different pressure

* Manufactured by the American Instrument Company, Washington, D. C. For a complete description see Rept. No. 107, National Advisory Committee for Aeronautics, 1921.

ranges, namely: a mercury manometer reading from — 12 lb. to + 12 lb. per sq. in. gage; and three Bourdon gages reading from 0 to 30 lb., from 15 to 200 lb., and from 100 to 1000 lb. per sq. in. gage, respectively.

The indicator diaphragm was sensitive to 0.1 lb. per sq. in. pressure, and as the mercury manometer was graduated to 0.1 lb. per sq. in., the determination of the low pressure parts of the cycle was made with corresponding accuracy. The Bourdon gages were graduated to pounds,* and calibrated against a dead weight tester before and during the series of tests. From a theoretical analysis of the motion of the indicator diaphragm it was determined that the greatest possible error in the pressure readings due to inertia of the diaphragm was 0.7 lb. per sq. in. for an engine running under the given test conditions. This error occurred in the part of the cycle immediately after ignition, at the time when the pressure was rising almost instantly to the maximum. For other parts of the cycle the error was much less than 0.7 lb. per sq. in., as the rate of pressure change was considerably less. Thus it is evident that the precision and accuracy of the indicator diaphragm is much greater than that of the gages on which the balancing pressures were read and that the indicator diagrams are correct to within 1 lb. per sq. in. at pressures above 30 lb. per sq. in. gage, and to 0.5 lb. per sq. in. at lower pressures.

41. *Method of Operation of Indicator.*—The high pressure air supply used for balancing the pressure in the cylinder was an air bottle containing air at a pressure of 800 lb. per sq. in. Control valves 1 and 2 on the air line were one-eighth inch brass needle valves fitted with convenient handles. Valve 1 controlled the admission of air to the indicator system and valve 2 the exhaustion of air by means of an aspirator.

Pressures were read for 59 different positions of the crank, as shown on the indicator log sheet of Test No. 22 (Table 2). The points at which pressures were read were so distributed that they fell close together at points of the cycle where very rapid pressure changes occurred.

In taking a pressure reading the balancing pressure was increased by admitting air through valve 1 until the clicks in the telephone receiver ceased. The pressure was then built up slightly and immediately decreased by exhausting air through valve 2 until the series of clicks started again. The mean of the pressures required to stop and start the series of clicks was taken as the actual pressure in the

* The 100-1000 lb. Bourdon gage was graduated to 10-pound intervals, but was not used except for a very few readings.

TABLE 2

INDICATOR LOG SHEET FOR TEST NO. 22

ENGINEERING EXPERIMENT STATION

UNIVERSITY OF ILLINOIS

SHEET FILE

Date July 26, 1922 Observer C. J. Rosecrans

Indicator Log SHEET

Test No. 22 Zero of Indicator before Test, +0.4 lb. per sq. in. Mean, +0.2 lb. per sq. in.
" " " after " 0 " " " "

Barometric Pressure, 14.4 lb. per sq. in.

No.	Time	Angle, degrees		Pressure lb. per sq. in. gage	Pressure lb. per sq. in. abs.	Piston Displacement per cent	Time	Angle, degrees		Pressure lb. per sq. in. gage	Pressure lb. per sq. in. abs.	Piston Displacement per cent
	1	2	3	4	5	6	7	8	9	10	11	
1	3:00 P.M.	0		52	66	0		180		+0.7	14.9	0
2		6		45	59	1.2		182		+0.9	15.1	0.2
3		8		41	55	2.3		190		1.4	15.6	3.5
4		10		38	52	3.5		200		2.0	16.2	14.0
5		12		35	49	5.0		210		2.0	16.2	29.4
6		17		27	41	10.0		220		1.3	15.5	47.0
7		21		20	34	15.2		230		0.4	14.6	64.4
8		24		16.5	30.7	19.8		240		0.3	14.5	79.5
9		30		10.0	24.2	29.4		250		1.5	15.7	90.8
10		36		4.9	19.1	40.0		260		3.6	17.8	97.7
11		42		2.2	16.4	50.4		270		7.2	21.4	100.0
12		48		-0.2	14.0	61.0		280		10.5	24.7	97.7
13		54		-2.1	12.1	70.7		290		13.5	27.7	90.8
14		60		-3.6	10.6	79.5		300		18.0	32.2	79.5
15		65		-4.7	9.5	85.8		307		23.0	37.2	69.4
16		70		-5.3	8.9	90.8		313		28.0	42.2	59.4
17		80		-6.1	8.1	97.7		318		36	50	50.4
18		90		-6.6	7.6	100.0		324		47	61	39.8
19		100		-6.9	7.3	97.7		330		63	77	29.4
20		110		-7.2	7.0	90.8		336		86	100	19.8
21		120		-7.7	6.5	79.5		338		101	115	16.7
22		130		-8.2	6.0	64.4		343		122	136	10.2
23		140		-8.2	6.0	47.0		345		132	146	8.0
24		150		-8.0	6.2	29.4		348		141	155	5.2
25		160		-6.6	7.6	14.0		350		145	159	3.5
26		168		-2.7	11.5	5.2		352		145	159	2.4
27		170		-1.6	12.6	3.5		353		115	129	1.9
28		173		-0.6	13.6	2.0		355		95	109	1.0
29		175		+0.2	14.4	1.0		360		52	66	0
30							3:25 P.M.	Max. Press.		147	161	

cylinder. Since the engine did not run with absolute regularity the clicks would sometimes occur intermittently when the balancing pressure and cylinder pressure were nearly in equilibrium. This irregularity occurred only at the time of maximum pressure and even then never covered a range of more than 4 lb. per sq. in. The mean of the pressures over which the clicks occurred intermittently was taken as

the actual cylinder pressure. However, during the greater part of the cycle a change in the balancing pressure of 1 lb. per sq. in. sufficed to stop or start the series of clicks, and for the low pressure parts of the cycle a much smaller pressure change produced the same effect.

The maximum pressure of the whole card was determined by short circuiting the indicator timer and balancing pressures as before.

42. *Method of Plotting Indicator Diagram.*—The indicator cards were plotted on the basis of piston displacement, giving a *PV* card. The piston displacements were determined graphically from the crank angles as read on the indicator timer, and were corrected for the angularity of the connecting rod. Corrections to the observed cylinder pressures were applied, as determined by the correction curves for the Bourdon gages.

Figure 3 shows the indicator diagram for Test No. 22, plotted from the data given on the log sheet (Table 2) after adding the barometric pressure, 14.4 lb. per sq. in. abs., and subtracting the "zero correction." The zero correction is the pressure required below the diaphragm to deflect the indicator diaphragm to touch the contact and give a click in the telephone receiver, with atmospheric pressure on the upper side of the diaphragm. It was never more than 0.5 lb. per sq. in. and was determined by setting the engine with the exhaust valve open and exhausting the space above the diaphragm until a click was heard in the receivers. The pressure difference above and below the diaphragm is the zero correction. It was determined before and after each test.

A low pressure indicator diagram (Fig. 1) was plotted in the same manner as the high pressure diagram, using a larger scale for pressures. From this card the data were taken for calculating the weight and analysis of the charge taken into the cylinder.

43. *Exhaust Gas Thermocouple.*—In order to determine the temperature of the exhaust gas left in the cylinder after the completion of the exhaust stroke, a thermocouple *T* (Fig. 8) was inserted in the exhaust valve cage.

This couple was constructed of No. 26 platinum—platinum-rhodium wire. The wires were welded in the oxy-gas flame and then flattened between two smooth steel plates to a thickness of about 0.0024 in. They were then filed down to a thickness of about 0.0015 in. and trimmed until the dimensions of a cross-section of the wire were approximately 0.008×0.0015 in. These dimensions were maintained for a distance of $\frac{1}{2}$ in. from the junction of the couple.

The couple was inserted through a two-hole porcelain tube into the exhaust valve cage, so that the couple junction was located $1\frac{1}{2}$ in. below the edge of the valve seat, and directly in the path of the exhaust gases.

The couple leads were soldered to copper wires, the soldered joints being used as cold junctions. These junctions were placed in a thermos flask and maintained at the temperature of melting ice. The couple was connected through a contact on the indicator timer to a Leeds and Northrup precision potentiometer.

The couple was calibrated in an electric furnace against a couple which had been standardized by the Bureau of Standards.

44. *Method of Using Exhaust Gas Thermocouple.*—The contact on the indicator timer for the thermocouple circuit was adjusted so that the couple circuit remained closed for a period of 20 degrees of crank angle, from 10 degrees before to 10 degrees after the dead center position at the end of the exhaust stroke.

A galvanometer having a period of $4\frac{1}{2}$ seconds was used with the potentiometer, and it was found that a balance could be obtained very readily with the circuit closed intermittently by the contact.*

It is, of course, evident that a couple of the dimensions given could not follow accurately all the variations of temperature caused by the exhaust gas. However, a temperature traverse of the whole cycle was undertaken, the results of which showed that for the latter part of the exhaust stroke the temperature was very nearly constant. Since the temperature desired was the temperature of the gases remaining in the clearance space at the end of the exhaust stroke, it is very probable that the thermocouple gave comparatively accurate results since the temperature remained constant for a large part of the cycle before the reading was actually taken. Moreover, an error in the exhaust temperature has little effect on the calculation of the weight of charge taken into the engine. Therefore the indications of the thermocouple may be regarded as correct within the limits of error of the other measurements.

45. *Fuel Gas Analysis.*—Inasmuch as the gas used for these tests was a complex mixture, a careful analysis of the gas for each test was essential. Analyses were made in a modified Orsat apparatus.† The carbon dioxide of the gas was absorbed by a solution

* A preliminary experiment was made by measuring a known temperature with a constantly closed circuit and an intermittently closed circuit. It was found that the temperature indications were the same, within the limits of error allowed. The frequency of the intermittent closing of the couple circuit was made commensurate with the speed of the engine.

† "Decomposition Processes Applicable to Certain Products of Coal Carbonization," M. J. Bradley with S. W. Parr. Chem. and Met. Eng., Vol. 27, pp. 737 to 744, 1922. Also S. W. Parr, "The Analysis of Fuel, Gas, Water, and Lubricants," 3rd. Ed., 1922.

of potassium hydroxide, the oxygen by potassium pyrogallate, the acetylene by ammoniacal silver chloride, the ethylene by bromine water, the aromatics by 20 per cent fuming sulphuric acid, the hydrogen and carbon monoxide were determined by combustion with copper oxide, the ethane and methane by slow combustion in pure oxygen, and the nitrogen was estimated by difference.

Samples of fuel gas were taken from the gasometer immediately following each test. All samples were taken over water in aspirator bottles. Analyses were made immediately after each test.

The results of the gas analysis for each test were based on at least two complete analyses. The percentage of the various constituents of the gas did not vary more than 0.5 per cent between the two analyses.

46. *Miscellaneous Measuring Apparatus.*—The ordinary mercury-in-glass laboratory thermometers were used for determining the water, gas, and air temperatures. The thermometers were calibrated against standard certified thermometers, and the corresponding corrections were applied to the readings.

The cooling water supplied to the engine was weighed in a large steel tank on platform scales. The scales were checked against standard weights throughout the range over which they were used and found to be correct and accurate.

A revolution counter applied to the engine shaft and dynamometer shaft was used for determining speeds.

A spring balance reading to 50 lb. was connected to the dynamometer for the purpose of obtaining the brake load. The scale was calibrated by dead weights, and corrections thus determined were applied to the readings.

In order to determine the position of the crank corresponding to ignition, the engine was turned over by hand until the timer made contact. The crank angle was then determined by measurement.

47. *Method of Conducting Test.*—In order to secure constant operating conditions for any given air-gas ratio a certain definite test procedure was followed for each test.

The engine was first started on gas from the main and run for at least 30 minutes, or until the exit temperature of the circulating water could be maintained nearly constant at 160 deg. F. During this period the gas and air flow were adjusted to give approximately the desired air-gas ratio.

At the end of the preliminary warming-up period the valve from the gas main was closed and the valve to gasometer 1 was opened simultaneously. The gas and air valves were then adjusted again to give the desired air-gas ratio. The time of ignition was set at various

points of the cycle, and the speed of the engine was determined for each point. The ignition was finally set at the point at which the engine ran at its highest speed, other conditions being maintained constant meanwhile. The engine was run under these conditions until the fuel in gasometer 1 was exhausted, and then the fuel supply was taken from gasometer 2.

As soon as the engine began drawing its fuel from gasometer 2, the recording of test data was started. The circulating water was turned into the weighing tank, the scale reading on the gasometer, together with the time, was noted and the operation of taking the indicator card was begun.

Observations of pressures, temperatures, speed, and other readings which remained practically constant were taken at 5-minute intervals. The recording of data for the indicator diagram occupied about 30 minutes. At the conclusion of this operation the gasometer scale was again read, the time noted, and the final readings of circulating water weights were taken. The exhaust temperature was next determined by means of the thermocouple, and then the engine was shut down.

Gas samples were taken immediately after the test from the remaining gas in gasometer 2. Humidity determinations of the air supplied were made by an Assman aspirating psychrometer hung directly above the tee in the top of the reservoir R (Fig. 7), from which the plug was removed.

48. *Sequence of Tests.*—A series of tests was run at a selected compression ratio, and then the compression ratio was changed and a new series was run. The test data are given in Table 3.

At times two tests were run at close intervals in order to avoid repeating the preliminary "warming-up" period. In such cases certain data, such as the barometric pressure and humidity, were recorded but once for the two tests. Gas samples were taken for both tests, but where the results of the analyses were almost identical, the average of the two analyses was used for the computation of the results.

The tests for which hydrogen was used as a fuel were all run at a compression ratio of 4.43. For these tests the hydrogen was piped from the ordinary commercial steel tanks into the gasometers from which the engine drew the gas.

TABLE 3

OBSERVED DATA

Test No.	Date	Compression Ratio	Mixture Ratio	Speeds—r. p. m.		Time of Weighing Cooling Water min. and sec.	Weight of Cooling Water lb.	Dynamometer Load lb.	Barometer in. Hg.	Orifice Head in. Water at 60 deg. F.	Static Pressure at Orifice in. Hg.
				Engine	Dynamometer						
7	6–28–22	4.00	7.789	336.2	2626	27–00	233	20.2	29.16	0.274	2.53
11	7– 7–22	4.00	5.308	333.0	2593	29–00	312	20.5	29.11	0.139	1.70
14	7–13–22	4.00	8.203	330.0	2577	30–30	258	20.3	29.52	0.262	2.45
16	7–19–22	4.00	10.558	324.8	2523	25–00	170	19.6	29.38	0.342	2.95
18	7–21–22	4.00	10.403	328.5	2562	32–00	220	19.6	29.28	0.340	2.94
19	7–21–22	4.00	8.720	333.7	2598	28–00	202	19.3	29.28	0.238	2.30
20	7–24–22	4.00	6.503	332.9	2613	30–30	308	19.0	29.36	0.149	1.75
21	7–24–22	4.00	5.106	337.3	2644	32–00	395	19.3	29.36	0.110	1.52
45	8–21–22	4.43	4.660	327.4	2581	28–30	321	19.6	29.28	0.108	1.50
46	8–21–22	4.43	6.394	328.3	2587	27–00	256	19.3	29.28	0.173	1.90
47	8–22–22	4.43	7.573	334.7	2656	27–00	220	19.5	29.20	0.238	2.30
48	8–22–22	4.43	8.808	323.6	2538	36–00	256	20.1	29.20	0.277	2.55
49	8–23–22	4.43	9.857	334.7	2663	32–30	216	19.3	29.15	0.319	2.80
50	8–23–22	4.43	9.044	336.8	2655	30–00	226	20.1	29.15	0.270	2.50
22	7–26–22	5.00	5.337	337.6	2630	30–00	326	19.2	29.30	0.105	1.48
23	7–26–22	5.00	6.981	335.7	2638	32–00	270	19.7	29.30	0.154	1.78
24	7–27–22	5.00	9.240	329.7	2595	36–30	256	19.7	29.33	0.236	2.12
25	7–27–22	5.00	11.173	330.0	2605	35–00	215	19.3	29.33	0.319	2.81
26	7–31–22	5.78	6.923	332.3	2604	33–00	261	19.6	29.28	0.147	1.73
27	7–31–22	5.78	8.381	331.1	2588	37–00	263	19.7	29.28	0.197	2.05
29	8– 1–22	5.78	10.842	330.2	2587	34–00	206	20.0	29.26	0.341	2.94
30	8– 3–22	5.78	10.347	335.0	2629	35–30	223	19.6	29.28	0.288	2.61
31	8– 3–22	5.78	11.734	324.1	2540	37–00	196	19.4	29.28	0.343	2.95
51 H	8–24–22	4.43	4.851	329.4	2578	23–30	166	19.6	29.05	0.271	2.50
52 H	8–24–22	4.43	3.921	331.8	2613	20–30	170	19.6	29.05	0.197	2.05
53 H	8–24–22	4.43	2.732	332.0	2608	24–30	262	19.2	29.05	0.117	1.55
54 H	8–25–22	4.43	5.759	329.7	2605	18–30	118	19.5	29.16	0.337	3.73
55 H	8–25–22	4.43	3.150	331.2	2603	19–00	219	20.1	29.16	0.148	1.75

TABLE 3

OBSERVED DATA (CONTINUED)

Test No.	Temperatures, deg. F.								Relative Humidity of Air to Engine	
	Wet Bulb	Dry Bulb	Room	Air at Orifice	Air at Engine	Gas at Engine	Cooling Water In	Cooling Water Out	Gasometer	Per Cent	At Deg. F.
7	60.6	82.4	87.4	94.2	87.2	110.1	60.4	160.0	87.8	27	82.4
11	61.7	81.1	80.1	89.4	83.0	103.7	60.3	160.8	79.0	32	81.1
14	66.6	86.0	87.5	93.7	87.8	108.0	61.8	154.6	88.7	36	86.0
16	64.6	83.1	82.9	91.8	83.9	105.6	60.2	158.4	85.5	37	83.1
18	67.8	87.6	89.9	93.8	88.4	107.1	68.4	159.6	89.7	36	87.6
19	67.8	87.6	90.6	97.1	90.0	108.7	61.7	159.3	90.9	36	87.6
20	66.0	87.8	86.5	96.3	88.2	108.4	62.8	157.8	85.9	31	87.8
21	66.0	87.8	88.0	98.7	89.6	115.9	61.7	151.8	86.7	31	87.8
45	65.1	85.5	83.9	93.2	85.8	107.5	62.1	159.8	82.5	33	85.5
46	65.1	85.5	85.3	94.0	87.1	107.6	61.3	148.0	83.1	33	85.5
47	65.7	85.6	82.0	89.0	83.5	101.6	62.2	156.3	81.0	35	85.6
48	65.7	85.6	86.4	93.2	86.5	105.2	63.1	156.1	87.1	35	85.6
49	62.8	81.9	79.2	89.4	82.4	100.8	65.7	160.0	79.1	34	81.9
50	62.8	81.9	79.5	90.8	83.0	102.2	65.5	154.0	79.2	34	81.9
22	68.0	89.2	90.1	96.5	90.8	114.1	61.0	155.0	89.3	33	89.2
23	68.0	89.2	90.4	96.6	91.2	112.5	61.6	159.0	88.7	33	89.2
24	69.1	90.9	93.9	96.7	92.3	109.2	62.4	155.8	94.0	33	90.9
25	69.1	90.9	94.1	97.3	92.5	110.8	62.0	157.0	91.4	33	90.9
26	67.1	88.2	91.1	95.0	89.6	106.2	61.6	161.3	89.4	33	88.2
27	67.1	88.2	91.2	95.5	90.4	106.8	61.3	159.6	87.4	33	88.2
29	66.4	88.3	91.4	95.7	90.2	107.9	60.0	154.8	90.4	31	88.3
30	64.6	83.7	81.5	90.1	82.5	96.0	60.3	157.0	81.2	35	83.7
31	64.6	83.7	84.3	91.2	84.5	102.7	60.0	164.0	82.9	35	83.7
51 H	66.2	89.1	90.4	92.7	88.5	110.8	64.2	159.8	87.1	29	89.1
52 H	66.2	89.1	91.0	95.2	90.2	118.0	64.0	160.1	85.2	29	89.1
53 H	66.2	89.1	91.3	96.7	91.3	122.5	63.2	162.0	85.0	29	89.1
54 H	64.0	85.5	83.4	90.0	85.8	103.9	64.1	157.3	82.8	30	85.5
55 H	64.0	85.5	84.5	93.4	86.8	110.9	61.6	148.8	80.6	30	85.5

TABLE 3

OBSERVED DATA (CONTINUED)

Test No.	Gasometer Reading inches		Time of Gasometer Reading min. and sec.	Exhaust Gas Temperature deg. F.	Consumption cu. ft. per min. at 60 deg. F. and 14.7 lb. per sq. in.		Heat of Combustion of Gas B. t. u. per standard cu. ft.	Heat of Combustion of Mixture B. t. u. per standard cu. ft.
	Initial	Final			Air	Gas		
7	7	62	26–39	1112	35.74	4.59	440.3	50.1
11	0	60	27–52	1208	25.71	4.84	447.8	71.0
14	0	54	30–08	885	35.16	4.29	456.2	49.6
16	3	55	30–24	1008	40.59	3.85	463.6	40.2
18	0	54	31–02	1040	40.28	3.87	438.3	42.3
19	2	47	26–10	1105	33.28	3.82	438.3	45.2
20	0	54	29–46	1200	26.48	4.07	464.0	61.8
21	0	59	29–21	1320	23.04	4.51	486.5	79.6
45	0	55	25–07	1330	23.00	4.94	461.4	81.5
46	0	52	26–23	1160	28.38	4.44	461.4	62.4
47	0	57	29–08	1090	33.39	4.41	443.1	51.7
48	13	57	23–54	1045	36.11	4.10	443.1	45.2
49	0	51	29–06	1125	39.02	3.96	478.7	44.1
50	0	52	28–49	1170	35.60	3.94	478.7	47.6
22	0	58	30–34	1020	22.56	4.23	481.0	75.8
23	0	50	29–01	1140	26.83	3.84	468.1	58.6
24	0	59	36–27	1145	33.05	3.58	446.9	43.6
25	0	54	34–25	1020	38.92	3.48	446.9	36.8
26	0	61	34–39	960	26.21	3.79	443.2	56.0
27	0	60	35–58	890	30.16	3.60	443.2	47.2
29	0	28	17–11	805	39.20	3.62	445.8	37.6
30	0	58	35–19	835	37.11	3.59	463.5	40.8
31	0	58	36–32	800	40.55	3.46	463.5	36.4
51 H	0	60	18–10	1054	35.54	7.33	256.5	42.8
52 H	0	52	15–09	1095	29.96	7.64	260.4	53.0
53 H	0	57	14–45	1305	23.51	8.61	258.3	69.2
54 H	0	54	17–07	1020	40.65	7.06	260.8	38.6
55 H	0	52	14–01	1155	26.32	8.36	260.8	62.8

TABLE 3

OBSERVED DATA (CONCLUDED)

Test No.	Analysis of Fuel Gas by Volume—Per Cent									
	CO_2	O_2	C_2H_2	C_2H_4	C_6H_6	H_2	CO	CH_4	C_2H_6	N_2
7	4.4	0.6	0.4	8.2	0.7	44.8	18.2	7.4	3.3	12.0
11	2.6	...	0.4	8.3	0.6	56.7	17.3	5.4	3.2	5.5
14	4.2	0.3	0.1	9.8	0.7	48.8	19.7	7.2	2.5	6.7
16	3.7	0.2	0.2	9.8	0.6	51.6	21.3	6.0	2.9	3.7
18	4.2	0.1	...	9.6	0.3	48.4	24.3	5.4	2.8	4.9
19	4.2	0.1	...	9.6	0.3	48.4	24.3	5.4	2.8	4.9
20	5.0	0.2	0.3	11.1	0.8	49.5	24.9	3.4	2.3	2.5
21	5.2	0.2	0.5	11.1	0.6	48.1	24.2	5.1	3.5	1.5
45	4.5	...	...	8.4	0.6	50.8	25.8	7.2	2.7	....
46	4.5	...	...	8.4	0.6	50.8	25.8	7.2	2.7	....
47	5.7	...	...	8.0	0.4	46.3	26.3	10.0	1.5	1.8
48	5.7	...	...	8.0	0.4	46.3	26.3	10.0	1.5	1.8
49	5.2	...	0.5	9.4	0.5	48.2	23.3	11.6	1.0	0.3
50	5.2	...	0.5	9.4	0.5	48.2	23.3	11.6	1.0	0.3
22	5.5	0.3	0.2	11.1	0.5	51.2	20.6	6.0	3.4	1.2
23	5.7	0.3	0.3	11.0	0.4	48.9	20.4	6.1	3.1	3.8
24	6.1	0.1	0.3	10.2	0.5	50.0	22.1	4.6	2.9	3.2
25	6.1	0.1	0.2	10.2	0.5	50.0	21.9	3.2	3.6	4.2
26	5.2	0.4	0.1	9.7	0.4	47.2	25.2	2.4	4.3	5.1
27	5.2	0.4	0.1	9.7	0.4	47.2	25.2	2.4	4.3	5.1
29	5.6	0.2	0.1	9.5	0.4	51.5	23.6	5.4	2.7	1.0
30	4.7	0.4	0.1	10.1	0.5	47.1	26.4	5.2	3.0	2.5
31	4.7	0.4	0.1	10.1	0.5	47.1	26.4	5.2	3.0	2.5
51 H	0.4	0.5	...	...	...	94.5	3.6	...	...	1.0
52 H	0.2	0.4	...	...	...	95.1	4.2	...	...	0.1
53 H	0.2	0.3	...	...	...	93.9	4.5	...	...	1.1
54 H	0.1	0.3	...	...	...	99.2	0.4	...	...	...
55 H	0.1	0.3	...	...	...	99.2	0.4	...	...	...

TABLE 4

CALCULATED RESULTS

Test No.	Compression Ratio	Mixture Ratio	H_v B.t.u. per mol at 60 deg. F.	Engine Speed r. p. m.	Ignition Degrees before Dead Center	Heat to Cooling Water per cent of H_v	Air cu. ft. per min. at 60 deg. F. and 14.7 lb. per sq. in.	Gas cu. ft. per min. at 60 deg. F. and 14.7 lb. per sq. in.	Temp. of Exhaust Gas deg. F. abs.	Pressure of Exhaust Gas lb. per sq. in. abs.
7	4.00	7.789	166 926	336.2	45	42.5	35.74	4.59	1620	14.5
11	4.00	5.308	169 773	332.8	30	49.7	25.71	4.84	1668	13.7
14	4.00	8.203	172 964	329.9	46	40.1	35.16	4.29	1345	15.2
16	4.00	10.558	175 737	324.8	60	37.3	40.59	3.85	1468	14.6
18	4.00	10.404	166 151	328.5	48	37.0	40.28	3.87	1500	15.2
19	4.00	8.720	166 151	333.7	27	42.0	33.28	3.82	1565	14.6
20	4.00	6.503	175 891	332.9	20	50.7	26.48	4.07	1660	14.2
21	4.00	5.106	184 440	337.3	15	50.6	23.04	4.51	1780	14.1
45	4.43	4.660	174 903	327.4	23	48.3	23.00	4.94	1790	14.1
46	4.43	6.394	174 903	328.3	25	40.2	28.38	4.44	1620	14.4
47	4.43	7.573	167 988	334.7	33	39.2	33.39	4.41	1550	14.6
48	4.43	8.808	167 988	324.2	38	36.4	36.11	4.10	1505	14.3
49	4.43	9.857	181 472	334.7	40	33.0	39.02	3.96	1585	14 9
50	4.43	9.044	181 472	336.8	35	35.4	35.60	3.94	1630	14.1
22	5.00	5.337	182 343	337.6	17	50.2	22.56	4.23	1480	14.9
23	5.00	6.982	177 462	335.7	20	45.6	26.83	3.84	1600	15.3
24	5.00	9.240	169 410	329.7	33	41.0	33.05	3.58	1605	14.9
25	5.00	11.173	169 063	330.0	50	37.6	38.92	3.48	1480	15.1
26	5.78	6.923	168 032	332.3	25	47.0	26.21	3.78	1420	14.5
27	5.78	8.381	168 032	331.1	30	43.8	30.16	3.60	1350	14.7
29	5.78	10.842	168 899	330.2	45	35.6	39.20	3.62	1265	15.5
30	5.78	10.347	175 714	335.0	37	36.5	37.10	3.59	1295	15.4
31	5.78	11.734	175 714	324.1	42	34.4	40.56	3.46	1260	15.2
51 H	4.43	4.851	97 251	329.4	15	35.8	35.54	7.33	1514	14.3
52 H	4.43	3.921	98 702	331.8	13	39.8	29.96	7.64	1555	14.1
53 H	4.43	2.732	97 993	332.0	5	47.5	23.51	8.61	1765	13.8
54 H	4.43	5.759	98 856	329.7	20	30.0	40.65	7.06	1480	14.6
55 H	4.43	3.150	98 856	331.2	10	46.1	26.32	8.36	1615	14.2

TABLE 4

CALCULATED RESULTS (CONTINUED)

Test No.	Analysis of Air by Volume Per Cent			Mol Composition of Charge within Cylinder			Mol Composition of Charge on Basis of 1 Mol of Gas			Sum of $CO+O_2+N_2$
	H_2O	O_2	N_2	Gas	Air	Exhaust Gas	Gas	Air	Exhaust Gas	
7	1.0	20.7	78.3	0.000 072 01	0.000 560 86	0.000 113 25	1.000	7.789	1.573	9.298
11	1.3	20.6	78.1	0.000 076 79	0.000 407 58	0.000 103 92	1.000	5.308	1.353	6.464
14	1.5	20.6	77.9	0.000 068 54	0.000 562 27	0.000 142 99	1.000	8.203	2.086	10.038
16	1.4	20.6	78.0	0.000 062 45	0.000 659 36	0.000 125 84	1.000	10.558	2.015	12.369
18	1.7	20.6	77.7	0.000 062 18	0.000 646 94	0.000 128 21	1.000	10.404	2.062	12.264
19	1.7	20.6	77.7	0.000 060 34	0.000 526 20	0.000 118 04	1.000	8.720	1.956	10.472
20	1.4	20.6	78.0	0.000 064 52	0.000 419 56	0.000 108 24	1.000	6.503	1.678	7.979
21	1.4	20.6	78.0	0.000 070 57	0.000 360 32	0.000 100 23	1.000	5.106	1.420	6.294
45	1.3	20.6	78.1	0.000 079 52	0.000 370 57	0.000 087 17	1.000	4.660	1.096	5.614
46	1.3	20.6	78.1	0.000 071 33	0.000 456 06	0.000 098 37	1.000	6.394	1.379	7.622
47	1.5	20.6	77 9	0.000 069 49	0.000 526 27	0.000 104 24	1.000	7.573	1.500	8.935
48	1.5	20.6	77.9	0.000 066 72	0.000 587 68	0.000 105 15	1.000	8.808	1.576	10.251
49	1.3	20.6	78.1	0.000 062 39	0.000 614 98	0.000 102 64	1.000	9.857	1.645	11.335
50	1.3	20.6	78.1	0.000 061 65	0.000 557 56	0.000 095 73	1.000	9.044	1.553	10.435
22	1.6	20.6	77.8	0.000 066 06	0.000 352 56	0.000 095 54	1.000	5.337	1.446	6.505
23	1.5	20.6	77.9	0.000 060 39	0.000 421 66	0.000 090 74	1.000	6.982	1.503	8.288
24	1.7	20.6	77.7	0.000 057 26	0.000 529 00	0.000 088 10	1.000	9.239	1.539	10.605
25	1.7	20.6	77.7	0.000 055 68	0.000 622 17	0.000 096 82	1.000	11.174	1.739	12.724
26	1.5	20.6	77.9	0.000 060 09	0.000 416 03	0.000 081 09	1.000	6.923	1.349	8.180
27	1.5	20.6	77.9	0.000 057 33	0.000 480 48	0.000 086 47	1.000	8.381	1.508	9.789
29	1.5	20.6	77.9	0.000 057 77	0.000 626 37	0.000 097 30	1.000	10.842	1.684	12.356
30	1.4	20.6	78.0	0.000 056 47	0.000 584 32	0.000 094 43	1.000	10.347	1.672	11.903
31	1.4	20.6	78.0	0.000 056 26	0.000 660 15	0.000 096 33	1.000	11.734	1.712	13.332
51 H	1.4	20.6	78.0	0.000 117 32	0.000 569 14	0.000 790 99	1.000	4.851	0.891	5.548
52 H	1.4	20.6	78.0	0.000 121 48	0.000 476 34	0.000 100 35	1.000	3.921	0.826	4.542
53 H	1.4	20.6	78.0	0.000 136 75	0.000 373 54	0.000 086 53	1.000	2.732	0.633	3.184
54 H	1.3	20.6	78.1	0.000 112 94	0.000 650 41	0.000 109 17	1.000	5.759	0.967	6.493
55 H	1.3	20.6	78.1	0.000 133 12	0.000 419 25	0.000 097 30	1.000	3.150	0.731	3.640

TABLE 4

CALCULATED RESULTS (CONTINUED)

Test No.	Analysis of Charge within Cylinder, in Mols											
	CO_2	H_2	CH_4	C_2H_2	C_2H_4	C_2H_6	C_6H_6	CO_2	H_2O	O_2	N_2	Total
7	0.175	0.431	0.071	0.004	0.079	0.032	0.007	0.146	0.294	1.759	7.364	10.362
11	0.167	0.548	0.053	0.004	0.080	0.030	0.006	0.140	0.336	1.145	5.152	7.661
14	0.188	0.466	0.069	0.001	0.093	0.024	0.007	0.175	0.416	1.883	7.967	11.289
16	0.204	0.493	0.057	0.002	0.094	0.028	0.006	0.139	0.385	2.408	9.757	13.573
18	0.231	0.461	0.051	.....	0.091	0.027	0.003	0.147	0.422	2.387	9.646	13.466
19	0.231	0.461	0.051	.....	0.091	0.027	0.003	0.160	0.411	1.997	8.244	11.676
20	0.239	0.474	0.032	0.003	0.106	0.022	0.008	0.195	0.362	1.450	6.290	9.181
21	0.231	0.460	0.049	0.005	0.106	0.033	0.006	0.211	0.362	1.084	4.979	7.526
45	0.248	0.488	0.069	.....	0.081	0.026	0.006	0.169	0.303	0.975	4.391	6.756
46	0.248	0.488	0.069	.....	0.081	0.026	0.006	0.162	0.319	1.402	5.972	8.773
47	0.252	0.445	0.096	.....	0.077	0.014	0.004	0.166	0.336	1.689	6.994	10.073
48	0.252	0.445	0.096	.....	0.077	0.014	0.004	0.156	0.341	1.975	8.024	11.384
49	0.225	0.466	0.112	0.005	0.091	0.009	0.005	0.148	0.331	2.204	8.906	12.502
50	0.225	0.466	0.112	0.005	0.091	0.009	0.005	0.151	0.323	2.013	8.197	11 597
22	0.196	0.489	0.057	0.002	0.106	0.032	0.005	0.200	0.387	1.143	5.166	7.783
23	0.194	0.466	0.058	0.003	0.105	0.030	0.004	0.173	0.358	1.546	6.548	9.485
24	0.209	0.472	0.044	0.003	0.096	0.027	0.005	0.150	0.377	2.067	8.329	11.779
25	0.208	0.475	0.030	0.002	0.097	0.034	0.005	0.145	0.400	2.515	10.001	13.912
26	0.240	0.450	0.023	0.001	0.093	0.041	0.004	0.158	0.322	1.534	6.406	9.272
27	0.240	0.450	0.023	0.001	0.093	0.041	0.004	0.151	0.337	1.877	7.672	10.889
29	0.224	0.489	0.051	0.001	0.090	0.026	0.004	0.139	0.370	2.438	9.694	13.526
30	0.254	0.453	0.050	0.001	0.097	0.029	0.005	0.141	0.340	2.325	9.324	13.019
31	0.254	0.453	0.050	0.001	0.097	0.029	0.005	0.132	0.347	2.633	10.445	14.446
51 H	0.034	0.904	.....	.....	.....	.....	.....	0.010	0.280	1.093	4.421	6.742
52 H	0.040	0.911	.....	.....	.....	.....	.....	0.010	0.284	0.875	3.627	5.747
53 H	0.043	0.900	.....	.....	.....	.....	.....	0.011	0.270	0.584	2.557	4.365
54 H	0.004	0.955	.....	.....	.....	.....	.....	0 002	0.276	1.298	5.191	7.726
55 H	0.004	0.955	.....	.....	.....	.....	.....	0.002	0.284	0.686	2.950	4.881

TABLE 4

CALCULATED RESULTS (CONTINUED)

Test No.	Analysis of Products of Complete Combustion, in Mols					Values of the Function $F = [H + (1 - z)H_v]$ Measured from 15 Per Cent on Compression Stroke to Various Points on Expansion Stroke				
						Per Cent of H_v at 520 deg. F. abs.				
	CO_2	H_2O	O_2	N_2	Total	2.5	5	7.5	10	15
7	0.664	1.146	0.903	7.364	10.077	24.98	23.66	25.04	25.76	26.77
11	0.624	1.262	0.282	5.152	7.320	40.72	36.53	35.52	35.65	36 59
14	0.710	1.300	1.000	7.967	10.977	27.16	25.47	25.25	25.63	27.22
16	0.684	1.284	1.516	9.757	13.241	36.56	33.03	30.60	28.80	26.55
18	0.683	1.257	1.549	9.646	13.135	43.41	29.95	27.89	25.99	23.62
19	0.696	1.246	1.159	8.244	11.345	34.08	24.29	22.27	20.85	18.42
20	0.776	1.205	0.567	6.290	8.838	21.55	18.77	18.04	18.22	18.94
21	0.815	1.254	0.150	4.979	7.198	29.01	25.09	24.48	23.67	23.63
45	0.736	1.187	0.090	4.391	6.404	28.13	27.42	26.29	26.07	26.36
46	0.729	1.203	0.517	5.972	8.421	40.24	23.86	22.82	21.58	20.82
47	0.720	1.181	0.839	6.994	9.734	35.59	26.63	21.82	21.39	21.06
48	0.710	1.186	1.125	8.024	11.045	28.96	25.54	23.55	22.15	21.77
49	0.725	1.250	1.280	8.906	12.161	46.74	38.64	32.14	27.62	24.52
50	0.728	1.242	1.089	8.197	11.256	37.34	30.15	24.26	19.58	16.16
22	0.763	1.315	0.213	5.166	7.457	39.89	29.43	23.98	23.93	24.67
23	0.725	1.255	0.642	6.548	9.170	52.35	29.73	24.05	21.16	16.62
24	0.685	1.228	1.211	8 329	11.453	31.90	23.53	17.83	15 20	14.36
25	0.679	1.248	1.660	10.001	13.588	46.49	35.41	27.24	22.06	16.91
26	0.715	1.140	0.688	6.406	8.949	42.85	28.13	21.14	20.51	19.38
27	0.708	1.155	1.031	7.672	10.566	42.67	28.83	19.05	15.82	14.89
29	0.672	1.232	1.586	9.694	13.184	47.71	33.48	23.03	18.12	13.48
30	0.729	1.190	1.439	9.324	12.682	51.65	40.40	33.19	27.46	19.26
31	0.720	1.197	1.747	10.445	14.109	54.42	45.74	39.14	33.67	25.71
51 H	0.044	1.184	0.624	4.421	6.273	49.46	37.41	28.93	22.99	20.55
52 H	0.050	1.195	0.400	3.627	5.272	33.82	22.63	21.21	21.17	21.94
53 H	0.054	1.170	0.113	2.557	3.894	23.27	23.56	24.36	24.98	27.54
54 H	0.006	1.231	0.818	5.191	7.246	50.15	39.37	30.26	24.85	21.00
55 H	0.006	1.239	0.206	2.950	4.401	25.35	22.78	23.27	23.90	25.24

TABLE 4

CALCULATED RESULTS (CONTINUED)

Test No.	Values of the Function $F = [H + (1-z)H_v]$ Measured from 15 Per Cent on Compression Stroke to Various Points on Expansion Stroke (continued)									Heat Loss 70 to 15 Per Cent Compression Stroke Per Cent of H_v at 520 deg. F. abs.
	Per Cent of H_v at 520 deg. F. abs.									
	20	30	40	50	60	70	80	90	100*	
7	26.63	28.54	29.49	29.22	28.69	27.43	25.31	23.04	20.50	3.02
11	37.90	40.66	43.28	43.71	41.86	41.22	41.12	41.24	41.80	2.27
14	27.61	27.35	27.23	27.42	27.74	27.18	27.15	27.64	28.42	2.00
16	25.22	24.13	25.99	26.39	26.03	24.26	23.78	23.55	23.60	2.53
18	22.75	21.13	21.63	22.97	23.03	22.64	21.19	20.49	20.50	2.91
19	17.35	17.69	19.13	18.35	18.13	17.93	18.51	18.53	18.08	3.11
20	20.46	23.66	25.31	26.23	25.61	25.66	26.00	26.25	26.83	2.21
21	24.24	24.71	24.45	25.61	26.89	29.03	29.50	29.90	30.00	2.50
45	26.97	28.46	30.53	31.63	32.54	32.95	33.35	33.63	33.72	1.31
46	20.97	23.66	26.05	28.09	29.22	29.72	29.67	29.29	28.60	1.26
47	21.52	21.69	21.42	23.77	25.23	26.48	27.33	28.61	29.80	1.54
48	22.01	21.21	20.37	19.80	20.21	21.06	22.03	23.61	25.42	1.98
49	22.70	23.08	22.01	21.43	22.24	23.20	24.08	25.64	26.65	1.56
50	15.61	16.83	18.16	19.87	20.82	22.80	25.29	29.23	25.50	0.52
22	24.96	24.76	25.22	25.80	27.00	27.45	27.86	28.69	29.10	3.24
23	15.63	16.42	18.83	19.27	19.29	19.40	19.52	19.70	19.80	4.40
24	14.56	14.94	15.51	15.82	16.16	16.20	16.14	16.98	18.20	4.18
25	14.06	13.10	14.64	15.59	15.44	15.50	15.13	14.73	14.20	3.10
26	19.80	20.67	21.90	23.03	23.06	22.54	21.37	20.80	19.60	3.22
27	14.79	17.60	18.00	18.62	18.02	18.70	18.62	18.51	18.03	3.79
29	11.69	12.06	12.35	13.80	13.78	13.92	13.12	12.22	11.20	4.93
30	14.43	11.02	11.99	13.68	13.99	12.72	13.18	12.70	11.92	4.67
31	20.86	16.70	15.69	15.34	14.88	14.57	13.77	13.79	13.10	4.45
51 H	20.38	21.77	21.86	21.58	22.30	22.72	23.88	25.73	28.78	0.42
52 H	23.58	24.25	26.30	26.99	27.87	28.83	29.43	30.62	32.20	1.05
53 H	28.26	29.31	31.19	32.20	33.07	33.72	35.43	37.62	40.30	0.66
54 H	19.56	19.85	21.01	21.78	23.19	23.84	24.43	25.36	26.30	1.20
55 H	26.00	26.79	29.46	30.89	32.17	33.50	34.53	36.07	37.90	1.67

*The values at 100% stroke were obtained from the curves.

TABLE 4

CALCULATED RESULTS (CONTINUED)

Test No.	Ideal Adiabatic Cycle												
	Adiabatic Compression				At End of Adiabatic Constant Volume Combustion				At End of Adiabatic Expansion				Thermal Efficiency of Complete Combustion Per Cent
	100 Per Cent Stroke		0 Per Cent Stroke										
	Temp. deg. F. abs.	Pressure lb. per sq. in. abs.	Temp. deg. F. abs.	Pressure lb. per sq. in. abs.	Temp. deg. F. abs.	Pressure lb. per sq. in. abs.	x	y	Temp. deg. F. abs.	Pressure lb. per sq. in. abs.	x	y	
7	783	11.55	1315	77.56	3867	221.8	1.0	1.0	2603	37.33	1.0	1.0	35.82
11	869	10.11	1441	67.04	4650	206.7	1.0	1.0	3315	36.84	1.0	1.0	33.07
14	736	11.27	1240	75.94	3700	220.3	1.0	1.0	2472	36.80	1.0	1.0	36.21
16	726	12.17	1230	82.46	3390	221.7	1.0	1.0	2212	36.17	1.0	1.0	37.14
18	745	12.32	1260	83.37	3324	214.5	1.0	1.0	2162	34.88	1.0	1.0	37.22
19	772	10.70	1300	72.07	3610	194.4	1.0	1.0	2395	32.25	1.0	1.0	36.33
20	790	9.29	1320	62.09	4230	191.5	1.0	1.0	2930	33.17	1.0	1.0	34.38
21	830	8.70	1370	57.41	4810	192.8	1.0	1.0	3480	34.86	1.0	1.0	32.29
45	799	8.76	1376	66.86	4972	229.0	1.0	1.0	3542	36.83	1.0	1.0	33.77
46	778	9.93	1351	76.41	4347	236.0	1.0	1.0	2947	36.12	1.0	1.0	36.19
47	759	10.85	1323	83.77	3941	241.1	1.0	1.0	2592	35.80	1.0	1.0	37.56
48	740	11.47	1294	88.89	3671	244.6	1.0	1.0	2366	35.60	1.0	1.0	38.67
49	735	11.67	1284	90.58	3645	250.1	1.0	1.0	2339	36.23	1.0	1.0	38.68
50	742	10.82	1327	85.73	3824	239.8	1.0	1.0	2485	35.17	1.0	1.0	37.51
22	779	8.45	1400	75.91	4722	245.3	1.0	1.0	3199	33.24	1.0	1.0	37.06
23	800	9.67	1450	87.59	4265	249.1	1.0	1.0	2765	32.30	1.0	1.0	39.06
24	722	10.26	1326	94.24	3654	252.5	1.0	1.0	2260	31.20	1.0	1.0	40.98
25	685	11.20	1268	103.60	3307	263.9	1.0	1.0	1990	31.80	1.0	1.0	42.02
26	730	8.88	1400	98.42	4170	282.9	1.0	1.0	2570	30.17	1.0	1.0	42.07
27	710	9.66	1370	107.80	3820	291.7	1.0	1.0	2280	30.09	1.0	1.0	43.55
29	670	11.42	1310	129.10	3390	325.6	1.0	1.0	1950	32.41	1.0	1.0	44.82
30	650	10.39	1260	116.40	3490	314.1	1.0	1.0	2030	31.61	1.0	1.0	44.56
31	660	11.69	1280	131.10	3320	332.1	1.0	1.0	1900	32.88	1.0	1.0	45.30
51 H	712	11.49	1275	91.12	3740	248.7	1.0	1.0	2413	36.22	1.0	1.0	38.28
52 H	760	10.84	1350	85.27	4160	241.0	1.0	1.0	2780	36.34	1.0	1.0	37.00
53 H	800	9.73	1430	77.06	4830	232.2	1.0	1.0	3430	37.22	1.0	1.0	34.01
54 H	700	12.47	1260	99.45	3500	259.1	1.0	1.0	2210	36.93	1.0	1.0	39.07
55 H	780	10.32	1400	82.10	4530	239.5	1.0	1.0	3130	37.36	1.0	1.0	36.46

TABLE 4

CALCULATED RESULTS (CONTINUED)

Test No.	At 70 Per Cent Compression Stroke			At 15 Per Cent Compression Stroke			Actual Cycle					
	Pressure lb. per sq. in. abs.	Volume cu. ft.	Temperature deg. F. abs.	Pressure lb. per sq. in. abs.	Volume cu. ft.	Temperature deg. F. abs.	Indicated Horse Power	Brake Horse Power	Mech. Efficiency Per Cent	Indicated Thermal Efficiency Per Cent	Brake Thermal Efficiency Per Cent	Indicated Thermal Efficiency of Compression, Explosion and Expansion Per Cent
7	16.4	0.4204	862	43.0	0.1966	1057	11.02	8.83	80.1	23.11	28.6	25.80
11	14.3	0.4204	953	37.0	0.1966	1153	10.41	8.86	85.1	20.35	17.3	24.08
14	16.0	0.4204	810	42.5	0.1966	1010	10.59	8.70	82.1	22.97	18.9	26.32
16	17.3	0.4204	800	46.2	0.1966	1000	10.18	8.25	81.0	24.14	19.6	26.94
18	17.5	0.4204	820	46.5	0.1966	1020	9.75	8.48	86.9	24.36	21.2	27.61
19	15.2	0.4204	850	40.2	0.1966	1040	9.62	8.36	86.9	24.36	21.2	28.54
20	13.2	0.4204	870	34.4	0.1966	1060	9.72	8.27	85.1	21.82	18.6	26.42
21	12.3	0.4204	910	30.9	0.1966	1070	10.52	8.51	80.8	20.34	16.4	24.12
45	12.6	0.4034	882	35.8	0.1796	1116	10.89	8.43	77.6	20.27	15.7	24.27
46	14.3	0.4034	860	41.3	0.1796	1105	11.07	8.32	75.7	22.76	17.2	27.01
47	15.6	0.4034	838	45.0	0.1796	1077	11.28	8.63	77.1	24.30	18.7	28.42
48	16.5	0.4034	817	48.5	0.1796	1070	11.03	8.50	77.0	25.65	19.8	29.58
49	16.8	0.4034	810	49.1	0.1796	1054	11.70	8.57	73.5	26.16	19.2	28.53
50	15.1	0.4034	795	45.8	0.1796	1073	11.41	8.89	78.0	25.66	20.0	29.67
22	12.3	0.3865	862	34.7	0.1627	1024	10.51	8.42	80.1	21.77	17.6	27.40
23	14.1	0.3865	887	39.4	0.1627	1044	10.62	8.66	81.5	25.04	20.4	30.38
24	15.0	0.3865	802	43.8	0.1627	986	10.65	8.52	80.0	28.24	22.6	33.23
25	16.4	0.3865	763	49.5	0.1627	970	10.93	8.38	76.7	29.84	22.9	34.25
26	13.1	0.3699	810	41.7	0.1461	1020	11.01	8.51	82.1	26.20	21.5	32.27
27	14.3	0.3699	790	45.4	0.1461	990	10.81	8.50	78.6	28.74	22.6	34.34
29	17.0	0.3699	750	54.0	0.1461	940	12.01	8.62	71.7	31.63	22.7	35.79
30	15.3	0.3699	720	48.2	0.1461	890	11.91	8.59	72.1	30.31	21.9	35.18
31	17.2	0.3699	730	54.9	0.1461	920	11.35	8.21	72.4	30.03	21.7	34.06
51 H	16.6	0.4034	790	50.5	0.1796	1070	11.15	8.42	75.5	25.18	19.0	28.81
52 H	15.6	0.4034	840	46.3	0.1796	1110	10.98	8.54	77.7	23.41	18.2	27.08
53 H	14.1	0.4034	890	42.0	0.1796	1180	10.76	8.35	77.5	20.55	16.0	24.15
54 H	18.1	0.4034	780	54.2	0.1796	1040	11.16	8.47	75.8	25.72	19.5	29.08
55 H	15.0	0.4034	870	43.5	0.1796	1120	11.05	8.72	78.9	21.50	17.0	25.12

TABLE 4

CALCULATED RESULTS (CONTINUED)

Test No.	Potential Efficiency Per Cent		Heat Balance Ideal Cycle		Heat Balance Actual Cycle Per Cent of H_V at 520 deg. F. abs.			
	Indicated Thermal Efficiency ÷ Ideal Efficiency	Indicated Thermal Efficiency of Compression Explosion and Expansion ÷ Ideal Efficiency	Work	Exhaust	Work	Pumping Losses	Heat Loss*	Exhaust Loss
			Per Cent of H_V at 520 deg. F. abs.					
7	64.51	72.05	35.82	64.18	25.8	2.6	23.5	48.1
11	61.53	72.82	33.07	66.93	24.1	5.7	44.1	26.1
14	63.43	72.09	36.21	63.79	26.3	3.4	30.4	39.9
16	65.00	72.57	37.14	52.86	26.9	2.8	26.1	44.2
18	65.71	74.18	37.22	62.78	27.6	2.2	23.4	46.8
19	67.05	78.56	36.33	63.67	28.5	4.2	21.2	46.1
20	63.47	76.85	34.38	65.62	26.4	4.6	29 0	40.0
21	62.99	74.70	32.29	67.71	24.1	3.8	32.5	39.6
45	60.02	71.87	33.77	66.23	24.3	4.0	35.0	36.7
46	62.95	74.71	36.19	63.81	27.0	4.2	29.9	38.9
47	64.70	75.66	37.56	62.44	28.4	4.1	31.3	36 2
48	66.33	76.52	38.67	61.33	29.6	3.9	27.4	39.1
49	67.63	73.76	38.68	61.32	28.5	2.4	28.2	40.9
50	68.46	79.16	37.51	62.49	29.7	4.0	26.0	40.3
22	58.74	73.93	37.06	62.94	27.4	5.6	32.4	34.6
23	64.21	77.80	39.06	60.94	30.4	5.3	24.5	39.8
24	68.90	81.09	40.98	59.02	33.2	5.0	19.8	42.0
25	71.00	81.53	42.02	57.98	34.2	4.4	17.3	44.1

*This heat loss is the sum of the heat loss from 70 per cent to 15 per cent on the compression stroke, and the loss from 15 per cent on compression stroke to 100 per cent on the expansion stroke.

TABLE 4

CALCULATED RESULTS (CONCLUDED)

Test No.	Potential Efficiency Per Cent		Heat Balance Ideal Cycle		Heat Balance Actual Cycle Per Cent of H_V at 520 deg. F. abs.			
	Indicated Thermal Efficiency ÷ Ideal Efficiency	Indicated Thermal Efficiency of Compression Explosion and Expansion ÷ Ideal Efficiency	Work	Exhaust	Work	Pumping Losses	Heat Loss*	Exhaust Loss
			Per Cent of H_V at 520 deg. F. abs.					
26	62.28	76.71	42.07	57.93	32.3	6.1	21.8	39.8
27	65.99	78.85	43.55	56.45	34.3	5.6	21.7	38.4
29	70.57	79.85	44.82	55.18	35.8	4.2	16.1	43.9
30	68.02	78.95	44.56	55.44	35.2	4.9	16.6	43.3
31	66.29	75.19	45.30	54.70	34.1	4.0	17.6	44.3
51 H	65.78	75.26	38.28	61.72	28.8	3.6	29.5	38.1
52 H	63.25	73.17	37.00	63.00	27.1	3.6	33.2	36.1
53 H	60.42	71.01	34.01	65.99	24.2	3.6	41.0	31.2
54 H	65.83	74.43	39.07	60.93	29.1	3.4	27.5	40.0
55 H	58.97	68.90	36.46	63.54	25.1	3.6	29.6	41.7

*This heat loss is the sum of the heat loss from 70 per cent to 15 per cent on the compression stroke, and the loss from 15 per cent on compression stroke to 100 per cent on the expansion stroke.

TABLE 5.—INDICATOR DIAGRAM DATA

Crank Angle Degrees	Piston Displacement Per Cent	Pressure in lb. per sq. in.									
		Test No. 7	Test No. 11	Test No. 14	Test No. 16	Test No. 18	Test No. 19	Test No. 20	Test No. 21	Test No. 45	Test No. 46
0	0.0	135.0	93.0	99.0	119.0	94.0	84.0	74.0	58.0	88.0	104.0
6	1.2	101.0	53 0	68.0	79.0	81.0	63.0	52.0	48.0	58.0	66.0
8	2.3	76.0	51.0	64.0	75.0	68.0	59.0	49.0	46.0	56.0	63.0
10	3.5	69.0	49.0	61.0	67.0	66.0	54.0	47.0	42.8	53.0	60.0
12	5.0	59.0	47.0	55.0	63.0	61.0	52.0	44.6	40.8	49.0	57.0
17	10.0	47.0	41.6	49.0	53.0	52.0	44.9	39.6	35.8	42.1	46.0
21	15.2	42.1	36.6	41.9	46.0	46.0	39.9	34.1	30.8	35 2	41.3
24	19.8	40.1	33.6	37.9	41.6	41.9	35.9	30.6	27.8	31.4	36.4
30	29.4	34.1	27.6	30.4	33.6	33.4	28.9	24.6	22.8	25.5	28.8
36	40.0	27.1	21 6	24.4	27.6	28.0	23.9	20.1	17.8	19.1	21.5
42	50.4	22.6	17.6	20.4	23.1	23.4	20.4	16.8	15.6	16.3	18.3
48	61.0	20.1	15.9	17.7	19.6	19.1	17.2	14.7	13.8	14.4	16.0
54	70.7	17.6	14.2	15.9	17.0	17.3	15.0	13.2	12.4	12.4	14.5
60	79.5	14.2	13.0	14.6	15.7	15.9	14.0	11.6	10.9	11.0	12.6
65	85.8	13.4	11.8	13.8	14.7	14.7	12.8	10.6	9.9	10.2	11.6
70	90.8	12.8	11.0	13.0	14.0	14.1	12.3	9.8	9.4	9.3	10.8
80	97.7	11.9	9.9	11.8	13 2	13.1	11.2	9.0	8.8	8.7	10.0
90	100.0	11.2	9.1	10.9	12.3	12.1	10.5	8.4	8.3	8.1	9.3
100	97.7	11.0	8.7	10.2	11.5	11.3	9.7	7.9	7.9	7.6	8.6
110	90.8	10.3	7.1	9.7	11.0	10.8	9.3	7.5	7.6	7.4	8.1
120	79.5	10.2	7.1	9.5	10.5	10.4	9.1	7.1	7.3	6.7	7.8
130	64.4	10.1	7.8	9.2	10.4	10.1	8.9	6.8	7.0	6.4	7.4
140	47.0	10.0	7.4	9.1	10.2	9.8	8.6	6.6	6.8	6.1	7.1
150	29.4	10.2	7.3	9.4	10.4	10.2	8.8	7.1	7.3	6.6	7.6
160	14.0	11.1	8.9	10.4	11.0	10.9	9.9	8.3	8.6	7.6	8.7
168	5.2	....	11.1	12.4	12.0	12.6	11.9	11.3	11.8	11.2	11.1
170	3.5	12.7	12.1	14.8	13.2	13.3	12.7	12.2	12.6	12.1	12.1
173	2.0	13.9	12.8	14.8	13.9	14.1	13.9	13.4	13.3	13.1	13.3
175	1.0	13.9	12.9	15.1	14.2	14.5	14.1	13.7	13.4	13.7	14.0
180	0.0	14.5	13.7	15.2	14.6	15.2	14.6	14.2	14.1	14.1	14.4
182	0.2	14.8	14.0	15.2	14.6	15.2	14.8	14.3	14.2	14.2	14.5
190	3.5	15.1	14.5	15.5	14.9	15.5	15.0	14.7	14.6	14.5	14.8
200	14.0	15.2	15.1	15.5	15.1	15.6	15.5	15.2	15.3	15.1	15.4
210	29.4	14.9	15.6	14.7	15.1	15.4	15.5	15.3	15.5	15.4	15.6
220	47.0	14.0	14.1	14.1	14.6	14.7	14.7	14.6	14.7	14.9	14.9
230	64.4	13.9	13.5	14.5	13.9	14.1	14.1	13.7	13.8	14.1	14.1
240	79.5	14.4	13.8	16.3	14.4	14.8	14.3	13.7	14.0	13.9	14.2
250	90.8	16.3	14.8	19.8	16.2	16.5	15.9	14.9	15.1	14.8	15.3
260	97.7	19.4	17.8	19.9	19.7	20.4	18.6	17.6	17.6	17.1	18.3
270	100.0	27.1	22.6	25.4	26.1	25.9	24.4	21.6	22.0	20.7	22.7
280	97.7	31.6	27.6	28.9	30.6	29.4	27.9	26.1	26.8	23.6	27.7
290	90.8	34.1	30.6	31.9	32.6	32.9	29.9	28.6	28.8	29.5	30.7
300	79.5	39.1	34.6	36.9	37.6	36.9	34.9	32.1	32.8	33.3	35.4
307	69.4	42.1	39.6	41.9	42.6	42.4	39.9	37.6	37.8	38.1	39.3
313	59.4	47.0	44.6	48.0	48.0	47.0	44.9	42.6	43.8	45.1	44.3
318	50.4	54.0	50.0	54.0	54.0	....	....	48.0	50.0	52.0	54.0
324	39.8	65.0	59.0	65.0	66.0	66.0	61.0	58.0	61.0	62.0	65.0
330	29.4	79.0	76.0	81.0	82.0	81.0	77.0	72.0	73.0	80.0	82.0
336	19.8	100.0	95.0	96.0	101.0	99.0	95.0	92.0	101.0	100.0	106.0
339	15.4	112.0	.....	.....	.....	.....	.....	.....	.....	113.0	120.0
343	10.2	130.0	124.0	133.0	130.0	129.0	121.0	118.0	119.0	130.0	139.0
345	8.0	140.0	133.0	142.0	137.0	137.0	128.0	128.0	126.0	142.0	160.0
348	5.2	155.0	144.0	153.0	145.0	141.0	133.0	138.0	132.0	158.0	162.0
350	3.5	166.0	151.0	161.0	151.0	148.0	139.0	147.0	142.0	164.0	166.0
351	3.0	168.0	152.0	162.0	153.0	149.0	139.0	.....	.....	.....	.....
352	2.4	169.0	152.0	165.0	157.0	.....	.....	149.0	142.0	170.0	167.0
353	1.9	170.0	153.0	.....	.....	124.0	131.0	.....	.....	173.0	163.0
354	1.4	170.0	153.0	.....	.....	.....	.....	.....	.....	.....	.....
355	1.0	172.0	153.0	163.0	155.0	114.0	104.0	129.0	74.0	160.0	140.0
360	0.0	146.0	94.0	99.0	119.0	94.0	82.0	76.0	50.0	.....	.....
Maximum	...	178.0	154.0	165.0	159.0	149.0	139.0	149.0	142.0	175.0	168.0

TABLE 5.—INDICATOR DIAGRAM DATA (CONTINUED)

Crank Angle Degrees	Piston Displacement Per Cent	Pressure in lb. per sq. in.								
		Test No. 47	Test No. 48	Test No. 49	Test No. 50	Test No. 22	Test No. 23	Test No. 24	Test No. 25	Test No. 26
0	0.0	110.0	133.0	102.0	109.0	66.0	80.0	104.0	124.0	104.0
6	1.2	76.0	91.0	84.0	83.0	59.0	67.0	79.0	89.0	77.0
8	2.3	71.0	80.0	80.0	74.0	55.0	63.0	72.0	82.0	71.0
10	3.5	66.0	75.0	74.0	67.0	52.0	59.0	67.0	76.0	66.0
12	5.0	63.0	66.0	70.0	64.0	49.0	55.0	63.0	72.0	62.0
17	10.0	52.0	56.0	57.0	55.0	41.0	46.0	52.0	58.0	50.0
21	15.2	44.3	49.0	50.0	47.0	34.0	40.0	43.1	50.0	41.1
24	19.8	39.3	42.3	43.1	39.9	30.7	34.6	38.6	43.1	36.1
30	29.4	31.6	34.5	35.2	33.1	24.2	27.6	32.1	33.1	28.1
36	40.0	25.6	27.7	28.6	26.3	19.1	22.6	24.1	27.1	22.1
42	50.4	19.7	21.3	21.7	20.1	16.4	18.1	21.1	22.1	17.4
48	61.0	17.1	18.3	18.8	17.0	14.0	15.8	16.8	18.2	14.9
54	70.7	15.6	16.4	16.6	14.9	12.1	14.1	14.8	16.2	13.1
60	79.5	14.0	14.6	15.2	13.7	10.6	12.4	13.7	14.7	11.2
65	85.8	12.9	13.9	14.0	12.9	9.5	11.3	12.4	13.7	10.1
70	90.8	12.3	13.0	13.6	11.7	8.9	10.6	11.5	12.6	9.4
80	97.7	10.8	12.1	12.5	11.1	8.1	9.8	10.4	11.6	8.5
90	100.0	10.0	10.8	11.3	9.9	7.6	9.3	9.7	10.6	8.0
100	97.7	9.3	10.1	10.8	8.7	7.3	8.8	9.0	9.8	7.5
110	90.8	8.9	9.5	10.1	8.7	7.0	8.4	8.6	9.4	7.0
120	79.5	8.5	9.3	9.5	8.7	6.5	7.6	8.3	9.2	6.6
130	64.4	8.4	9.0	9.1	8.6	6.0	7.3	7.9	9.1	6.2
140	47.0	8.1	8.8	9.1	8.7	6.0	7.3	7.9	9.1	6.2
150	29.4	8.3	9.2	9.5	9.1	6.2	7.4	7.1	9.4	6.5
160	14.0	9.3	10.2	10.1	10.2	7.6	8.6	9.0	10.3	7.3
168	5.2	11.5	11.8	12.0	11.9	11.5	12.3	11.9	12.0	10.4
170	3.5	12.2	12.3	12.4	12.6	12.6	13.1	12.9	13.2	11.9
173	2.0	13.5	13.2	13.7	13.7	13.6	14.4	14.1	14.3	13.4
175	1.0	14.3	13.8	14.3	14.0	14.4	14.8	14.4	14.6	14.2
180	0.0	14.6	14.3	14.9	14.1	14.9	15.3	14.9	15.1	14.5
182	0.2	14.8	14.4	15.0	14.3	15.1	15.4	15.0	15.2	14.6
190	3.5	15.0	14.7	15.4	15.1	15.6	15.8	15.2	15.4	15.0
200	14.0	15.3	15.2	15.7	15.3	16.2	16.3	15.5	15.4	15.3
210	29.4	15.1	15.4	15.4	15.4	16.2	16.3	15.3	15.0	15.5
220	47.0	14.4	14.8	14.7	15.2	15.5	15.3	14.5	14.4	14.8
230	64.4	14.0	14.0	14.1	14.2	14.6	14.7	14.1	14.2	14.1
240	79.5	14.3	14.2	14.4	13.9	14.5	14.9	14.4	14.5	14.1
250	90.8	15.8	15.5	16.1	14.2	15.7	16.2	15.8	16.6	15.2
260	97.7	19.2	19.0	19.6	15.9	17.8	18.8	18.9	20.3	17.6
270	100.0	24.4	23.8	23.8	19.0	21.4	22.6	22.6	26.1	21.1
280	97.7	27.7	29.7	29.5	23.4	24.7	26.6	27.1	29.1	25.1
290	90.8	30.7	32.6	32.4	29.3	27.7	29.1	29.1	32.1	28.1
300	79.5	35.4	37.4	38.1	35.2	32.2	34.1	34.1	36.1	32.1
307	69.4	42.3	43.3	44.1	41.9	37.2	39.0	40.1	42.1	36.1
313	59.4	46.0	51.0	51.0	50.0	42.2	45.0	45.1	48.0	43.1
318	50.4	56.0	58.0	59.0	57.0	50.2	52.0	50.0	56.0	47.0
324	39.8	70.0	70.0	70.0	70.0	61.0	64.0	65.0	69.0	61.0
330	29.4	84.0	87.0	88.0	87.0	77.0	82.0	82.0	88.0	80.0
336	19.8	108.0	108.0	110.0	110.0	100.0	105.0	107.0	112.0	104.0
339	15.4	122.0	123.0	124.0	126.0	115.0	115.0	.....	.....	122.0
343	10.2	141.0	145.0	141.0	144.0	136.0	140.0	145.0	144.0	150.0
345	8.0	153.0	155.0	149.0	150.0	146.0	145.0	157.0	154.0	160.0
348	5.2	163.0	166.0	154.0	156.0	155.0	157.0	166.0	161.0	169.0
350	3.5	165.0	177.0	156.0	163.0	159.0	157.0	173.0	164.0	177.0
351	3.0	.....	.....	.....	.....	.....	.....	.....	.....	.....
352	2.4	168.0	182.0	160.0	166.0	159.0	137.0	176.0	165.0	174.0
353	1.9	164.0	181.0	159.0	165.0	129.0	130.0	161.0	164.0	168.0
354	1.4	.....	.....	.....	.....	.....	.....	.....	.....	.....
355	1.0	151.0	174.0	152.0	150.0	109.0	105.0	139.0	154.0	144.0
360	0.0	111.0	133.0	102.0	109.0	66.0	80.0	104.0	124.0	104.0
Maximum	...	168.0	184.0	161.0	170.0	159.0	158.0	176.0	166.0	178.0

TABLE 5.—INDICATOR DIAGRAM DATA (CONCLUDED)

Crank Angle Degrees	Piston Displacement Per Cent	Pressure in lb. per sq. in.								
		Test No. 27	Test No. 29	Test No. 30	Test No. 31	Test No. 51H	Test No. 52H	Test No. 53H	Test No. 54H	Test No. 55H
0	0.0	114.0	165.0	119.0	144.0	88.0	83.0	70.0	101.0	75.0
6	1.2	86.0	121.0	97.0	109.0	81.0	73.0	66.0	85.0	69.0
8	2.3	79.0	104.0	88.0	99.0	76.0	70.0	63.0	82.0	66.0
10	3.5	75.0	90.0	80.0	89.0	73.0	68.0	60.0	79.0	63.0
12	5.0	71.0	83.0	72.0	82.0	69.0	63.0	57.0	74.0	60.0
17	10.0	56.0	65.0	59.0	66.0	58.0	54.0	49.0	62.0	51.0
21	15.2	44.1	53.0	48.0	55.0	50.0	45.2	41.1	54.0	42.1
24	19.8	38.1	47.0	41.2	47.0	43.0	40.1	37.2	47.0	38.1
30	29.4	30.1	37.1	32.2	36.1	35.1	33.3	29.5	39.1	30.5
36	40.0	24.1	29.2	26.2	28.1	28.4	26.4	21.9	30.5	24.4
42	50.4	19.1	23.7	20.2	24.1	21.2	20.1	18.6	25.4	19.1
48	61.0	16.1	19.7	17.2	19.6	18.5	17.3	15.8	20.1	16.6
54	70.7	14.2	16.8	15.2	17.1	16.5	15.6	14.1	17.8	14.9
60	79.5	12.7	15.3	13.9	15.3	14.7	13.6	12.4	16.2	13.5
65	85.8	11.5	14.4	12.5	14.3	14.0	13.2	11.3	15.2	12.3
70	90.8	10.6	13.8	11.6	13.7	13.5	12.4	10.5	14.4	11.5
80	97.7	9.7	12.4	10.7	12.6	12.5	11.4	9.7	13.7	10.6
90	100.0	8.9	11.4	9.7	11.6	11.8	10.6	9.1	13.0	10.1
100	97.7	8.4	10.5	9.1	10.7	11.0	10.1	8.9	12.4	9.7
110	90.8	7.9	9.9	8.8	10.1	10.5	9.8	8.6	11.9	9.4
120	79.5	7.6	9.8	8.5	9.8	10.0	9.1	8.1	11.2	8.8
130	64.4	7.1	9.5	8.3	9.6	9.0	8.5	7.3	10.3	8.1
140	47.0	7.0	9.3	8.2	9.3	8.5	7.4	6.7	9.7	7.3
150	29.4	7.3	9.7	8.3	9.6	9.0	8.1	7.3	9.6	7.8
160	14.0	8.0	10.4	9.2	10.3	9.6	9.2	8.3	9.8	9.0
168	5.2	11.1	12.8	11.8	12.6	11.4	11.4	11.2	11.8	11.3
170	3.5	12.3	13.7	13.0	13.4	12.3	11.9	12.2	12.6	12.3
173	2.0	13.8	14.6	14.2	14.3	13.2	13.1	13.1	13.6	13.3
175	1.0	14.3	15.0	14.7	14.7	13.7	13.6	13.5	14.0	13.8
180	0.0	14.7	15.5	15.4	15.2	14.3	14.1	13.8	14.6	14.2
182	0.2	14.9	15.6	15.4	15.2	14.3	14.2	13.9	14.7	14.2
190	3.5	15.1	15.7	15.5	15.4	14.7	14.6	14.2	15.1	14.6
200	11.0	15.4	15.6	15.3	15.6	15.2	15.2	14.7	15.4	15.1
210	29.4	15.4	15.3	14.8	15.3	15.3	15.4	15.5	15.5	15.7
220	47.0	14.9	14.5	14.2	14.4	14.7	14.1	15.2	14.9	15.3
230	64.4	14.1	14.4	14.2	14.1	13.8	13.8	14.1	14.1	14.3
240	79.5	14.3	15.0	14.9	14.9	14.0	13.9	13.6	14.3	13.9
250	90.8	15.7	16.9	16.7	16.9	15.5	15.1	14.4	16.1	14.7
260	97.7	18.2	20.5	20.4	21.2	19.0	17.9	16.6	19.6	17.2
270	100.0	21.6	26.2	26.2	28.1	23.5	22.3	20.6	24.4	21.3
280	97.7	26.1	29.1	29.2	31.1	29.4	27.5	25.9	30.5	24.7
290	90.8	28.1	33.1	32.2	33.1	32.3	30.5	28.7	33.3	29.6
300	79.5	33.1	37.1	37.2	38.1	38.0	35.2	34.3	38.1	34.3
307	69.4	38.1	43.1	42.2	44.1	43.0	41.1	39.1	45.2	40.1
313	59.4	44.1	50.0	49.0	49.0	52.0	47.0	45.2	52.0	46.0
318	50.4	52.0	58.0	57.0	61.0	59.0	56.0	53.0	60.0	55.0
324	39.8	63.0	74.0	72.0	75.0	72.0	67.0	63.0	74.0	66.0
330	29.4	84.0	94.0	92.0	92.0	90.0	85.0	81.0	93.0	84.0
336	19.8	111.0	126.0	121.0	123.0	113.0	107.0	102.0	117.0	106.0
339	15.4	129.0	145.0	138.0	138.0	130.0	122.0	115.0	132.0	120.0
343	10.2	157.0	168.0	152.0	150.0	148.0	143.0	137.0	152.0	143.0
345	8.0	170.0	184.0	161.0	163.0	154.0	156.0	149.0	159.0	154.0
348	5.2	178.0	190.0	169.0	172.0	158.0	171.0	166.0	161.0	.....
350	3.5	180.0	192.0	172.0	182.0	160.0	172.0	174.0	162.0	181.0
351	3.0	.....	.....	.....	.....	.....	.....	.....	.....	.....
352	2.4	181.0	194.0	179.0	184.0	156.0	169.0	183.0	161.0	185.0
353	1.9	179.0	191.0	178.0	178.0	153.0	158.0	.....	.....	186.0
354	1.4	.....	.....	.....	.....	.....	.....	.....	.....	.....
355	1.0	154.0	184.0	144.0	164.0	114.0	140.0	119.0	133.0	114.0
360	0.0	114.0	164.0	119.0	144.0	90.0	83.0	71.0	101.0	75.0
Maximum	...	181.0	194.0	181.0	181.0	160.0	173.0	185.0	162.0	187.0

V. Discussion of Results

49. *Tables of Results.*—The results of the tests are given in Tables 3, 4, and 5. Table 3 contains the observed data and Table 4 the calculated results. Table 5 contains the indicator diagram data for all tests.

50. *Effect of Air-Gas Ratio on Thermal Efficiency.*—In Fig. 12 are shown the thermal efficiencies obtained when illuminating gas was used as a fuel and the air-gas ratio varied. The engine had a compression ratio of 5.00. Three thermal efficiency curves are shown:

(1) the ideal thermal efficiency of the adiabatic cycle, based on the assumption that complete combustion had occurred at the time maximum pressure was attained;

(2) the thermal efficiency of compression, explosion, and expansion, based on the actual indicator diagram;

(3) the indicated thermal efficiency.

It is evident from Fig. 12 that the method of calculating the ideal cycle gives an efficiency curve of very nearly the same shape as the actual efficiency curves, and lying practically parallel to them. The ideal efficiencies range from 37.1 per cent to 42.0 per cent while the actual efficiencies range from 21.9 per cent to 29.9 per cent. This difference is, of course, due to the heat loss during the cycle and the work of suction and exhaust. The curve of thermal efficiency of compression, explosion, and expansion lies between the ideal and the actual curves. The difference between the curve of thermal efficiency of compression, explosion, and expansion and the curve of indicated thermal efficiency (of the whole cycle) is, of course, due to the loss occasioned by the work of suction and exhaust. The difference between the ideal (adiabatic) efficiency curve and the curve of thermal efficiency of compression, explosion, and expansion is due to the slow combustion and the heat loss during compression, explosion, and expansion. It is evident, then, that the method of calculation employed in these tests gives a means of separating the "pumping" losses from the heat losses.

It should be noted that the ideal thermal efficiencies given by the upper curve are calculated on the basis of complete combustion. Test No. 22, at a mixture ratio of 5.337, was also calculated assuming dissociation (see pages 32 to 33, and the resulting ideal thermal efficiency was 36.76 per cent, as compared with 37.06 per cent for the complete combustion case. For Test No. 22 the value of x at equilibrium was 0.9288, and the value of y was 0.988. It is evident that

for this particular gaseous fuel the difference in the efficiencies calculated on the basis of complete and incomplete combustion respectively is small. For the other tests of this series, at a compression ratio of 5.00, the values of x and y at equilibrium were found to be unity (or practically so). That is, with leaner mixtures, dissociation becomes less, and for all practical purposes it can be assumed that x and y are unity.

By similar calculations and comparison it was found that the dissociation at other compression ratios was small, and therefore the ideal efficiency curves shown are those based on calculations assuming complete combustion at the time of maximum temperature.

Figs. 10, 11, and 13 show the three efficiency curves for tests in which the engine was operated with compression ratios of 4.00, 4.43, and 5.78, respectively. These curves show the same general characteristics as those previously discussed. Fig. 14 shows the thermal efficiencies obtained when hydrogen was used as a fuel, the engine operating at a compression ratio of 4.43. The curves have the same general appearance as those for illuminating gas.

Fig. 15 shows the ideal and indicated thermal efficiencies plotted against the heat of combustion of one cubic foot* of the mixture charged into the engine. The air standard efficiencies calculated from the formula

$$E = 1 - \left(\frac{1}{r}\right)^{K-1}$$

are also shown. In this formula r is the compression ratio, and K is the ratio of the specific heats, or 1.4.

It is evident from these curves that the curve for ideal thermal efficiency continued back to the ordinate of zero heat of combustion (or to the point where the mixture is pure air) coincides very well with the air-standard efficiency. The curves of indicated thermal efficiency have almost the same slope as the ideal efficiency curves and intersect the zero ordinate at a certain distance below the ideal efficiency curves. This distance represents the heat loss which would occur if the actual engine with its accompanying heat losses was operated under the conditions adopted for the definition of the air-standard cycle.

51. *Effect of Compression Ratio on Thermal Efficiency.*—Fig. 16 shows the effect of compression ratio on the indicated thermal efficiency, for various air-gas ratios. These curves were derived from the efficiency curves described in Section 50.

* The cubic foot referred to is the cubic foot mxture at 60 deg. F. and 29.92 in. of mercury pressure.

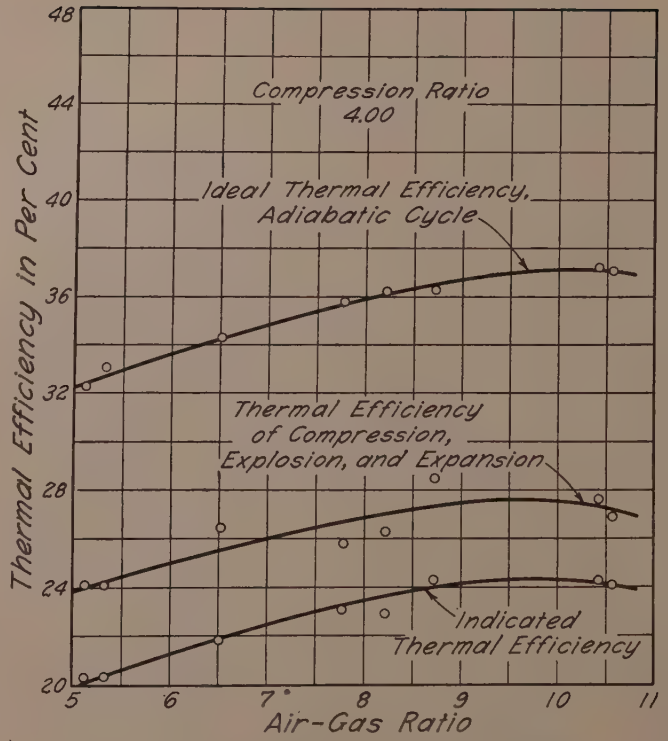

FIG. 10. THERMAL EFFICIENCIES FOR TESTS AT COMPRESSION RATIO 4.00

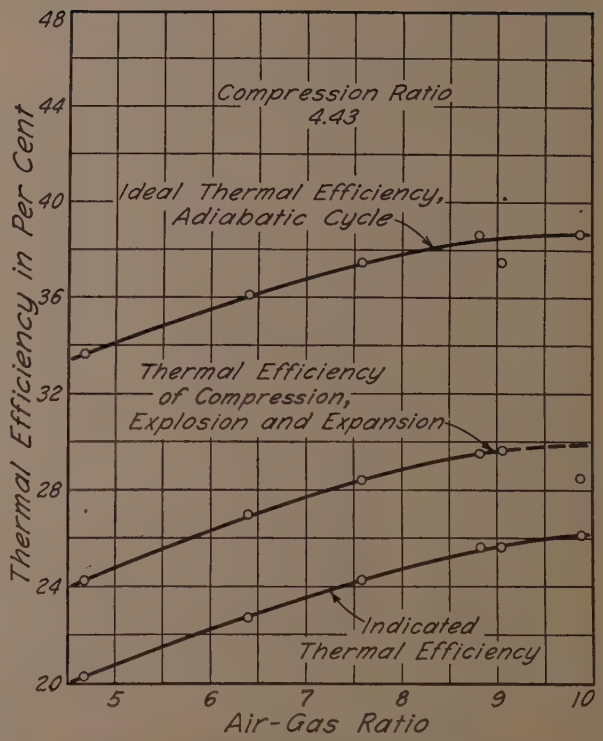

FIG. 11. THERMAL EFFICIENCIES FOR TESTS AT COMPRESSION RATIO 4.43

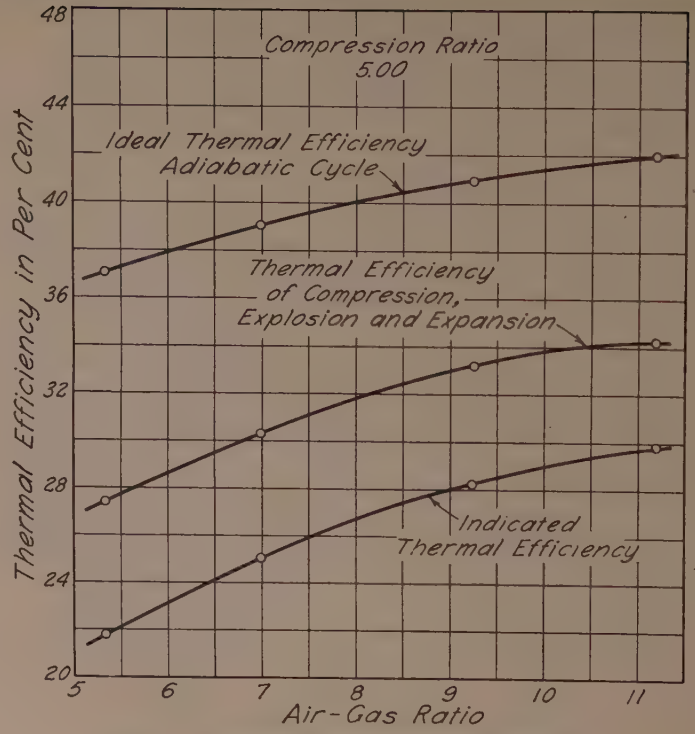

FIG. 12. THERMAL EFFICIENCIES FOR TESTS AT COMPRESSION RATIO 5.00

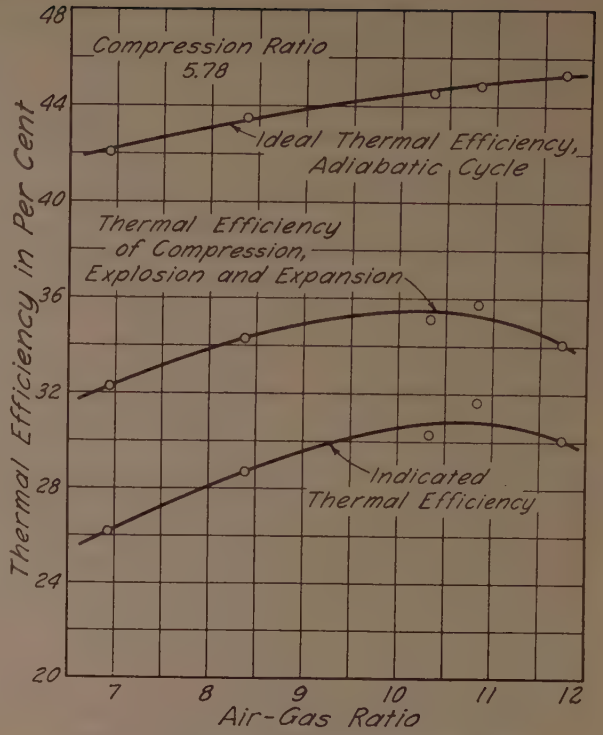

FIG. 13. THERMAL EFFICIENCIES FOR TESTS AT COMPRESSION RATIO 5.78

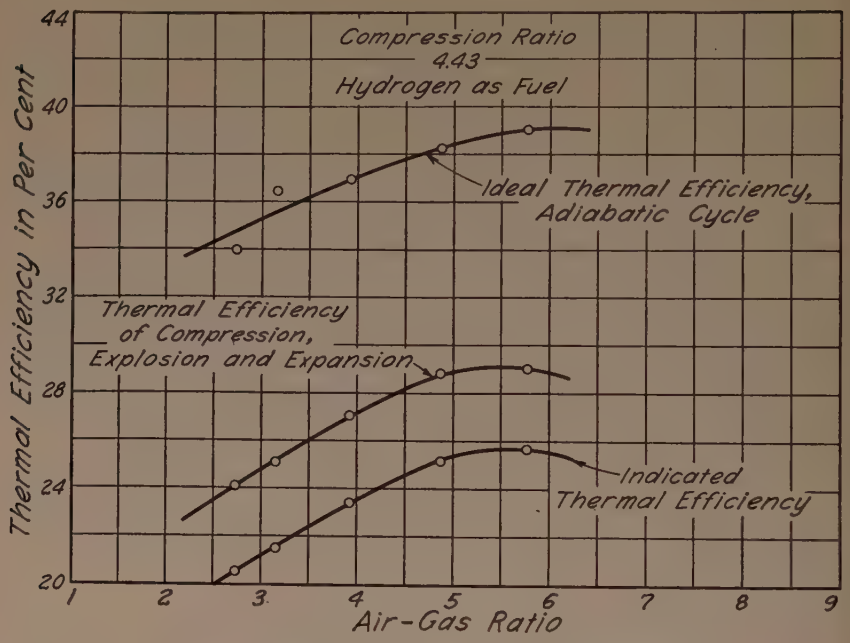

FIG. 14. THERMAL EFFICIENCES FOR TESTS AT COMPRESSION RATIO 4.43, USING HYDROGEN AS FUEL

From these curves it is evident that in general an increase in compression ratio results in an increase in efficiency.

This increase in efficiency is partly due to the fact that when higher compression ratios are employed, the superficial area of the combustion space at the time of ignition is reduced, thus reducing the heat loss; also the maximum gas temperature is raised, and the amount of work from the cycle is thereby increased. The greater ratio of expansion results in a lower temperature at the end of expansion, thus tending to increase the efficiency.

Fig. 16 also shows the ideal thermal efficiencies, plotted against compression ratio, for various air-gas ratios. It may be noted that the slope of the indicated efficiency curves at the lower compression ratios is practically the same as the slope of the ideal efficiency curves. At higher compression ratios, however, the slope of the indicated thermal efficiency curves decreases, while the ideal efficiency curves continue as practically straight lines. From the increasing difference in slope of the two sets of curves the influence of the rapidly increasing heat losses at higher compression is at once evident.

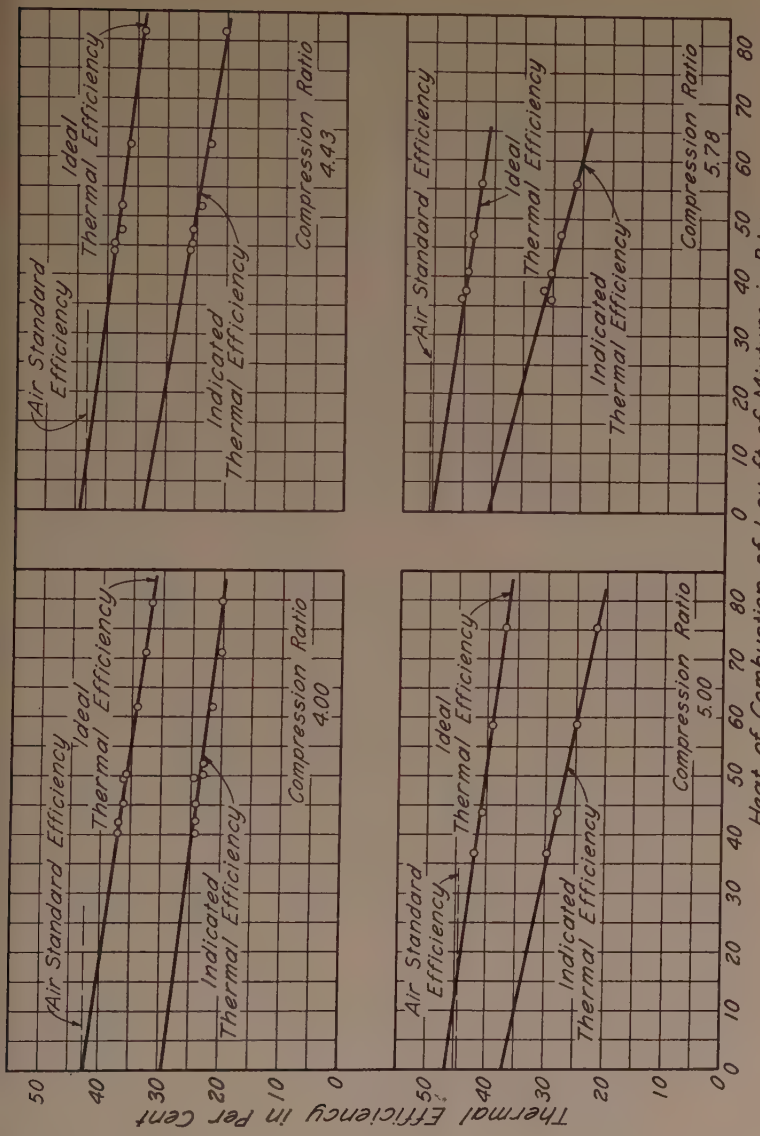
Thermal Efficiency in Per Cent
Heat of Combustion of 1 cu. ft. of Mixture in B.t.u.
Air Standard Efficiency
Ideal Thermal Efficiency
Indicated Thermal Efficiency
Compression Ratio 4.43
Compression Ratio 4.00
Compression Ratio 5.78
Compression Ratio 5.00

FIG. 15. EFFECT OF HEAT OF COMBUSTION OF MIXTURE ON THERMAL EFFICIENCY FOR TESTS AT VARIOUS COMPRESSION RATIOS

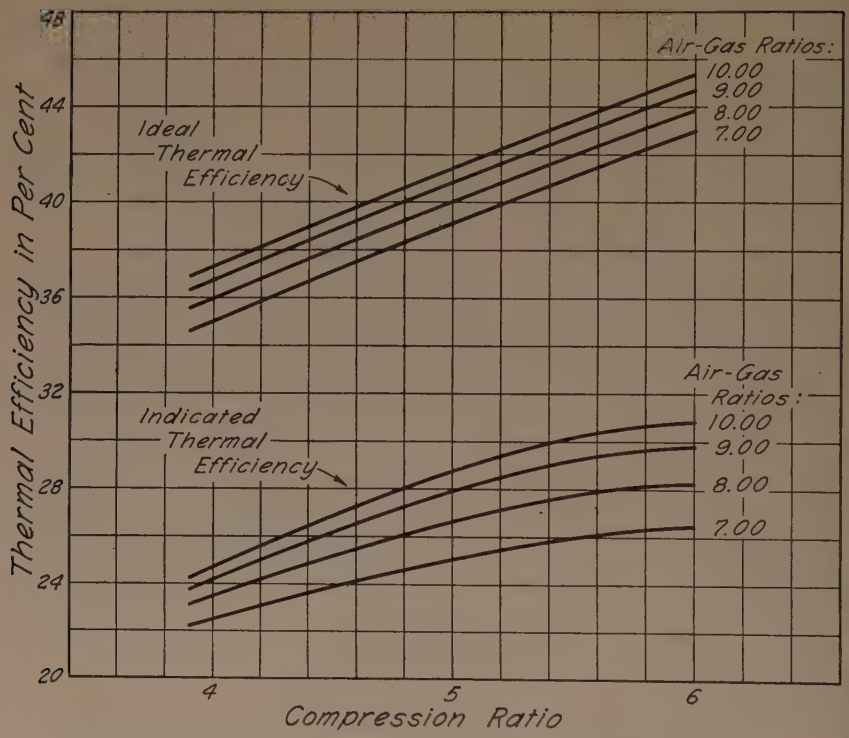

FIG. 16. EFFECT OF COMPRESSION RATIO ON THERMAL EFFICIENCY

A few tests were also made on the engine adjusted for a compression ratio of 6.50. These tests show a lower indicated thermal efficiency than those at a compression ratio of 5.78.

It was very noticeable, however, that the engine ran irregularly when the 6.50 compression ratio was used and that while no definite evidence of detonation or other abnormal phenomena was obtained, the operation of the engine was not as smooth as it was when lower compression ratios were employed.

For this reason the tests for the 6.50 compression ratio are not given in full. The decrease in thermal efficiency was noticeable, and can unquestionably be attributed to the generally unsatisfactory operation of the engine.

52. *Effect of Different Fuels on Thermal Efficiency.*—In Fig. 17 the ideal and indicated thermal efficiencies for tests at a compression ratio of 4.43, and with illuminating gas and hydrogen as fuels are shown. It is evident from these curves that the efficiencies, both ideal

and actual, using the two fuels, cover approximately the same range. The tests with hydrogen were made with lower air-gas ratios, as was demanded by the characteristics of the fuel.

It is evident, however, that a greater difference might have been observed between results obtained with the two fuels if the illuminating gas had not contained so large a proportion of hydrogen, thus making its action quite similar to that of the hydrogen fuel itself.

The apparent discrepancy in the two points on the ideal efficiency curve for hydrogen (Fig. 17) for air-gas ratios of 2.73 and 3.15 is explained by the difference in the amount of impurities in the hydrogen as drawn from the gasometer (see Table 3).

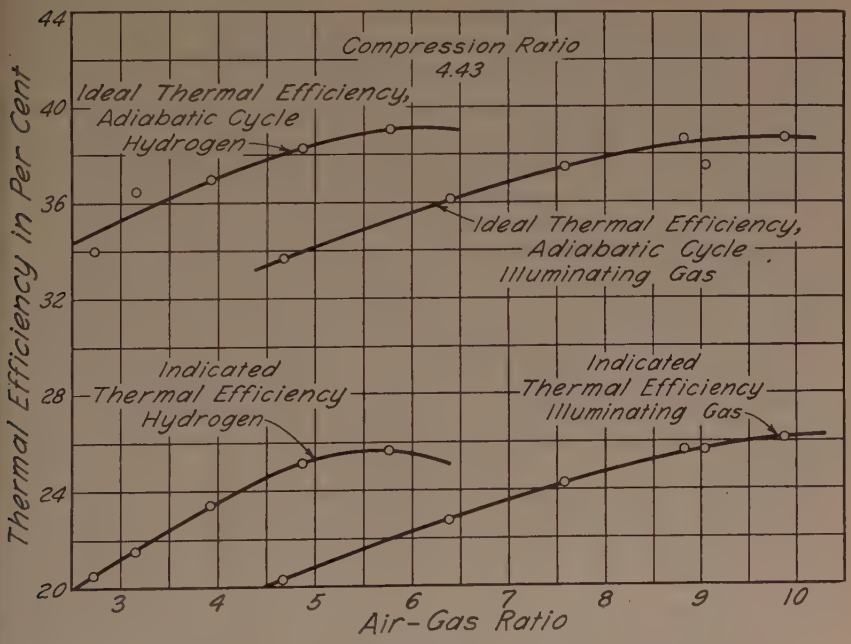

FIG. 17. EFFECT OF DIFFERENT FUELS ON THERMAL EFFICIENCY

53. *Potential Efficiency.*—From the ideal efficiencies and the efficiencies of compression, explosion, and expansion a potential efficiency curve has been computed for tests at a compression ratio of 5.00 (see Fig. 18). The potential efficiency increases with an increase in air-gas ratio since the heat loss for the weaker mixtures is relatively lower. Any factor that will increase the velocity of reaction or reduce the heat loss will increase the potential efficiency: e.g., dual ignition, turbulence, etc.

The potential efficiency of the compression, explosion, and expansion process tends to reach a maximum and then decrease as the mixture is further weakened, since the rate of combustion becomes so low that a portion of the unburned gas escapes into the exhaust. The potential efficiency based on the net actual indicated efficiency does not show this maximum since a greater total weight of charge has to be drawn in when the weaker mixtures are used for a given load and thus the work of suction is reduced.

Potential efficiency curves for the tests at other compression ratios are also shown in Fig. 18. These curves exhibit the same general characteristics as those already discussed.

54. *Heat Loss and Reaction Velocity.*—In Fig. 21 are plotted the curves of the function $[H + (1 - z)H_r]$ for tests at a compression ratio of 5.00. The method of calculating this function was described in Section 25. H is the heat loss to the walls of the cylinder from 15 per cent of the compression stroke to any given point on the expansion stroke; z is the progress of the explosive reaction, considered as a whole; H_r is the heat of combustion of the fuel for conditions existing at 15 per cent of the compression stroke.

Taking the curve of this function for Test No. 22 (Fig. 21), it is at once evident that at 0 per cent of the stroke (compression dead center) the value of $[H + (1 - z)H_r]$ is very high. At this time H is zero, since inflammation has not yet occurred, also z is zero. Hence the value of the function is simply H_r. As the piston moves out, however, the reaction progresses, and thus the factor $(1 - z)H_v$ decreases. At the same time H, the heat loss, increases. The heat loss does not increase as rapidly as the factor $(1 - z)H_v$ decreases, and so at first the value of the function decreases, as shown in Fig. 21. After a short time, however, the temperature of the gases is high, and the heat loss increases rapidly, so that its effect balances the decrease in the factor $(1 - z)H_v$, and the curve attains a minimum (at 9 per cent stroke for Test No. 22). At this point the rate of heat loss equals the rate of evolution of heat by the reaction, and maximum pressure also exists. Then the heat loss increases, and the value of $(1 - z)H_v$ decreases slowly (since the reaction is very near completion). Therefore the curve rises, and gradually attains a maximum near the end of the stroke.

Let the function $[H + (1 - z)H_v]$ be denoted by F

$$F = H + (1 - z)H_v$$

Differentiating with respect to time

$$\frac{dF}{dt} = \frac{dH}{dt} - H_v \frac{dz}{dt}$$

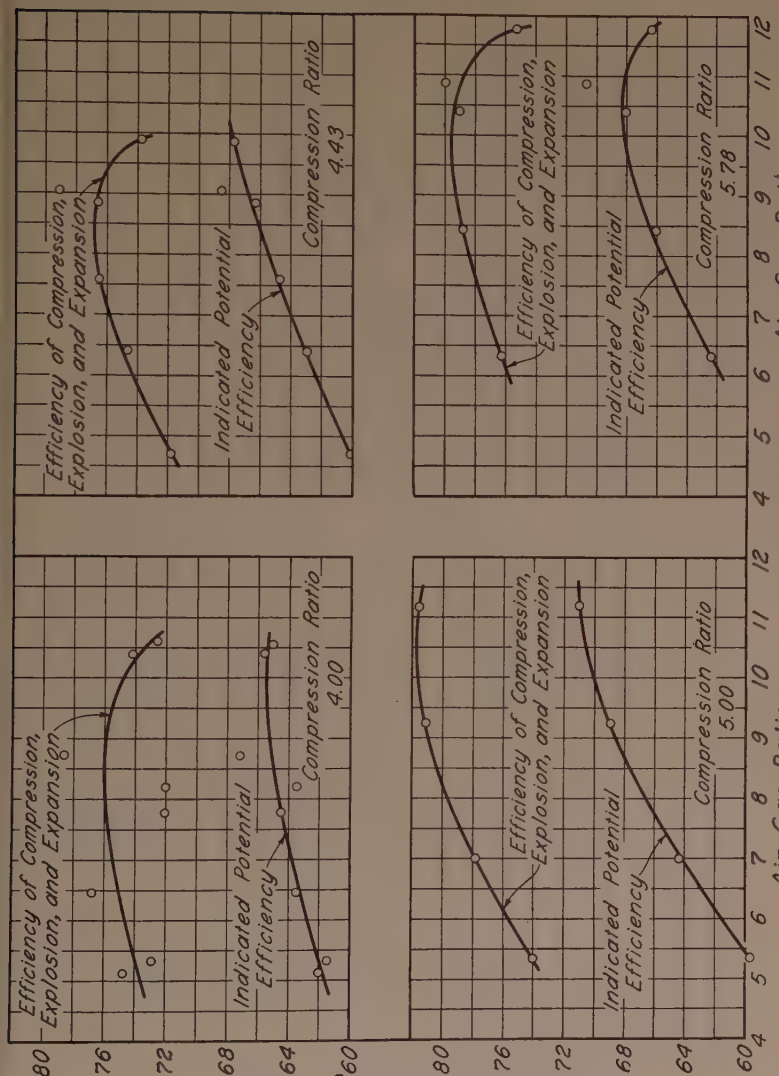

FIG. 18. POTENTIAL EFFICIENCY FOR TESTS AT VARIOUS COMPRESSION RATIOS

in which $\frac{dH}{dt}$ is the rate of heat loss and $\frac{dz}{dt}$ is a measure of the reaction velocity. It is highly probable that $\frac{dH}{dt}$ is always positive; that is, the flow of heat is always from the hot gases to the cooler cylinder walls. It is hardly probable that the gas temperatures, even at the end of expansion, become less than the temperature of the cylinder walls.

The following analysis substantiates the hypothesis that $\frac{dH}{dt}$ is always positive. During the most active part of combustion, that is, during the period of attainment of maximum temperature, the piston is near the dead center so that the only surfaces exposed to the gas are the black carbonized surfaces of the piston and the cylinder head. These surfaces also have a fairly low temperature, due to the cooling effect of the incoming charge, and therefore form ideal surfaces for absorbing radiation. A material decrease in the temperature of the gases close to the walls is therefore effected during the combustion. As the polished surface of the cylinder wall is uncovered no great increase in heat loss can be expected from the increase in the area exposed since the polished surface will reflect a considerable amount of the incident radiation. During the expansion stroke the gas temperature decreases and a slow conduction of heat back through the metal from the hot black surfaces also occurs. The mean gas temperature dropped about 950 deg. F. during the expansion for Test No. 24 and about 730 deg. F. for Test No. 25. This material and rapid drop in temperature may be sufficient to lower the gas temperature below that of the temperature of the black surfaces (this temperature being maintained because of the low rate of conduction) so that radiation of heat back to the gases occurs during the latter part of the expansion stroke.

Measurement of heat losses made on explosions in closed vessels shows that a decrease in the rate of total heat loss occurs as the time elapsed after ignition increases. In the actual engine cylinder it would be expected that the effect of increased area during the expansion stroke would counteract the effect of decreased temperature so that the rate of heat loss would tend to be more or less constant. This is found to be the case for Tests Nos. 22 and 23.

For Test No. 22 the temperature drop during expansion is 1230 deg. F. and for Test No. 23 it is 980 deg. F. The maximum temperatures are 3720 and 3330 deg. F. abs., respectively. For Tests Nos.

24 and 25 the maximum temperatures are 2950 and 2580 deg. F. abs., respectively. The maximum temperatures for Nos. 22 and 23 are greater than for Tests Nos. 24 and 25. Since the rate of radiation is a function of T^n, where n is probably about 4; the temperature of the carbon surface should be much higher for Tests Nos. 22 and 23 than for Nos. 24 and 25. The first two tests show a greater drop in temperature during expansion than the last two tests. It would therefore be reasonable to expect some indication of radiation back to the gas from the walls during the latter part of the expansion stroke for Nos. 22 and 23 but the curves of $[H + (1 - z)H_v]$ do not show any such radiation.

A calculation was made for Tests Nos. 24 and 25 to determine if the drop in the $[H + (1 - z)H_v]$ curves was due to combustion of the lubricating oil used. From this calculation it was found that the combustion of the lubricating oil would account for only a very small part of the drop in the curves. In general, it may therefore be concluded that $\frac{dH}{dt}$ is a positive quantity, and if the function $\frac{dF}{dt}$ (which represents the slope of the "F" curve) becomes negative, it follows immediately that $\frac{dz}{dt}$ must be positive; that is, chemical energy is being liberated by some reaction taking place. Fig. 21 shows the curves of the function F for tests with a compression ratio of 5.00.

It may be noted that for Test No. 23 the function F, after passing through a minimum, continues to increase steadily till the end of the expansion stroke is reached. This curve thus indicates that the reaction has been completed early in the stroke, and that no unburned gases, which will burn later in the stroke, are left in the valve pockets and other recesses.

The same reasoning applies for Test No. 22, except that a second period of reaction, or perhaps a more intense phase of reaction, occurs between 20 and 35 per cent stroke, as shown by the negative slope of the F curve.

The $[H + (1 - z)H_v]$ curves for Tests Nos. 24 and 25 show a secondary addition of heat to the gases during the latter period of the expansion stroke after the initial combustion is apparently complete. This reduction in the function $[H + (1 - z)H_v]$ may be explained by the fact that the weak mixtures are comparatively slow burning. The flame of the initial combustion may not reach the gases in the

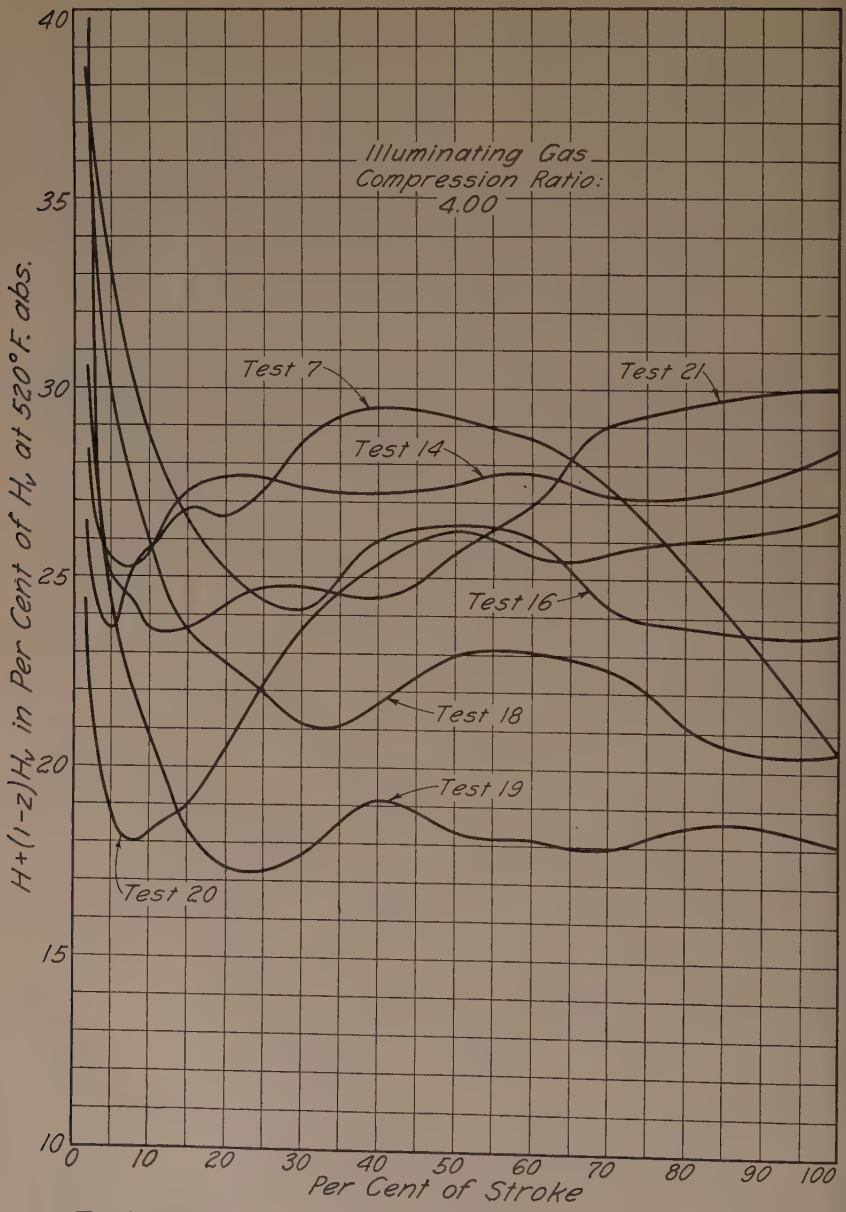

FIG. 19. CURVES OF THE FUNCTION $[H+(1-z)H_v]$ FOR TESTS AT COMPRESSION RATIO 4.00

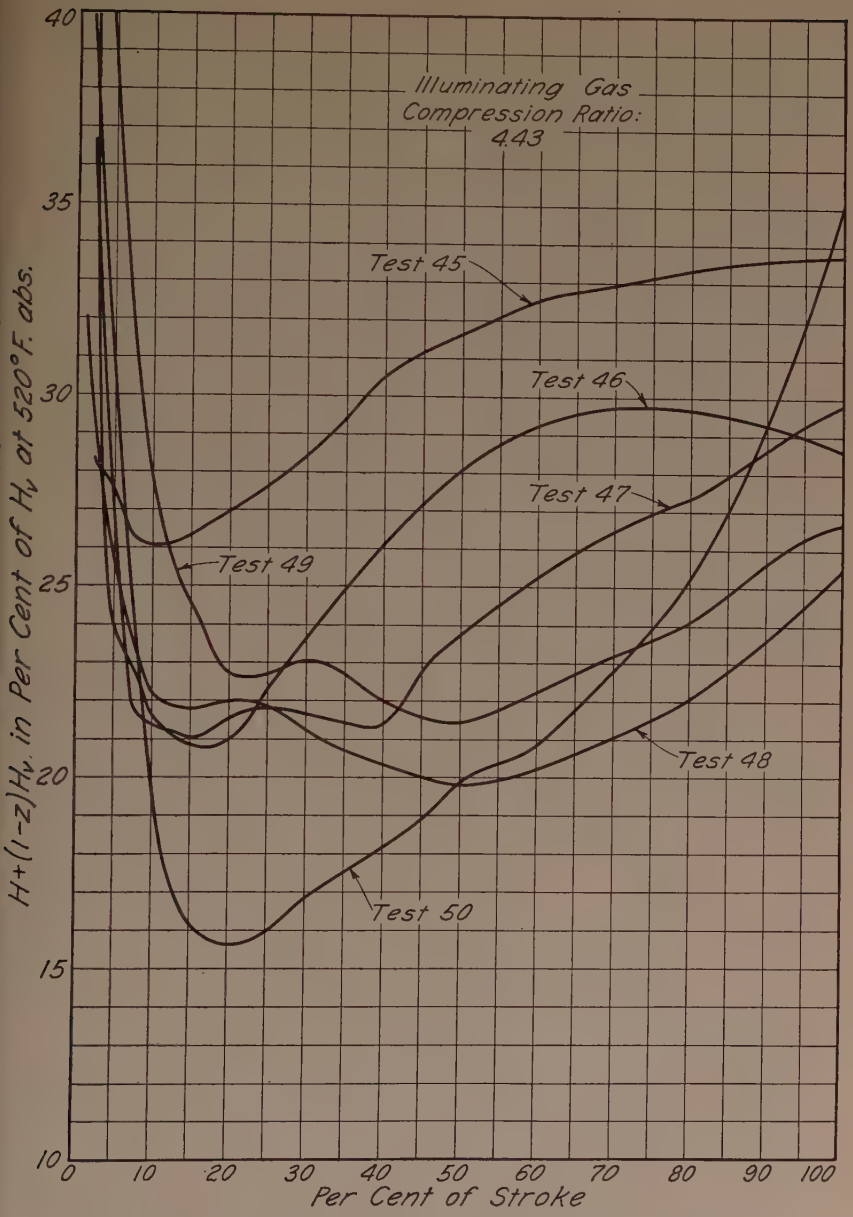

FIG. 20. CURVES OF THE FUNCTION $[H+(1-z)H_v]$ FOR TESTS AT COMPRESSION RATIO 4.43

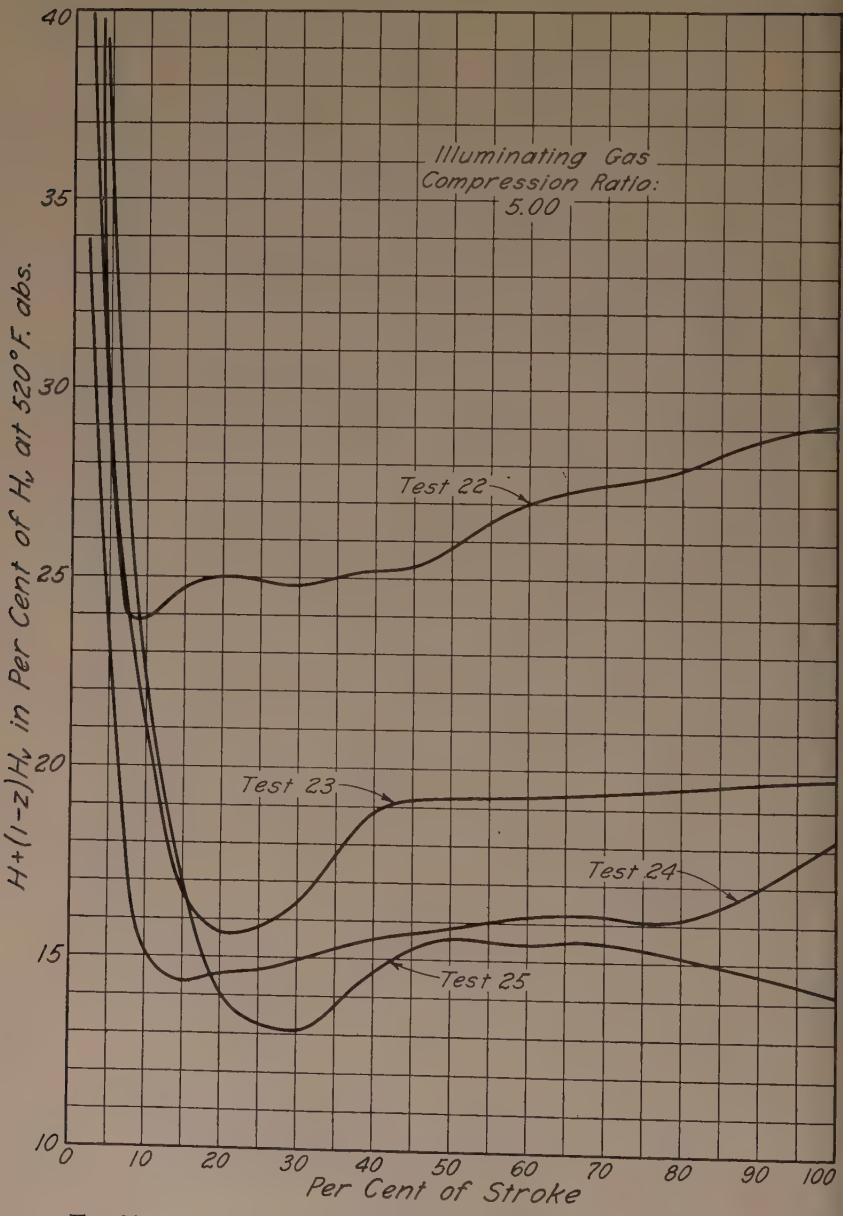

FIG. 21. CURVES OF THE FUNCTION $[H+(1-z)H_v]$ FOR TESTS AT COMPRESSION RATIO 5.00

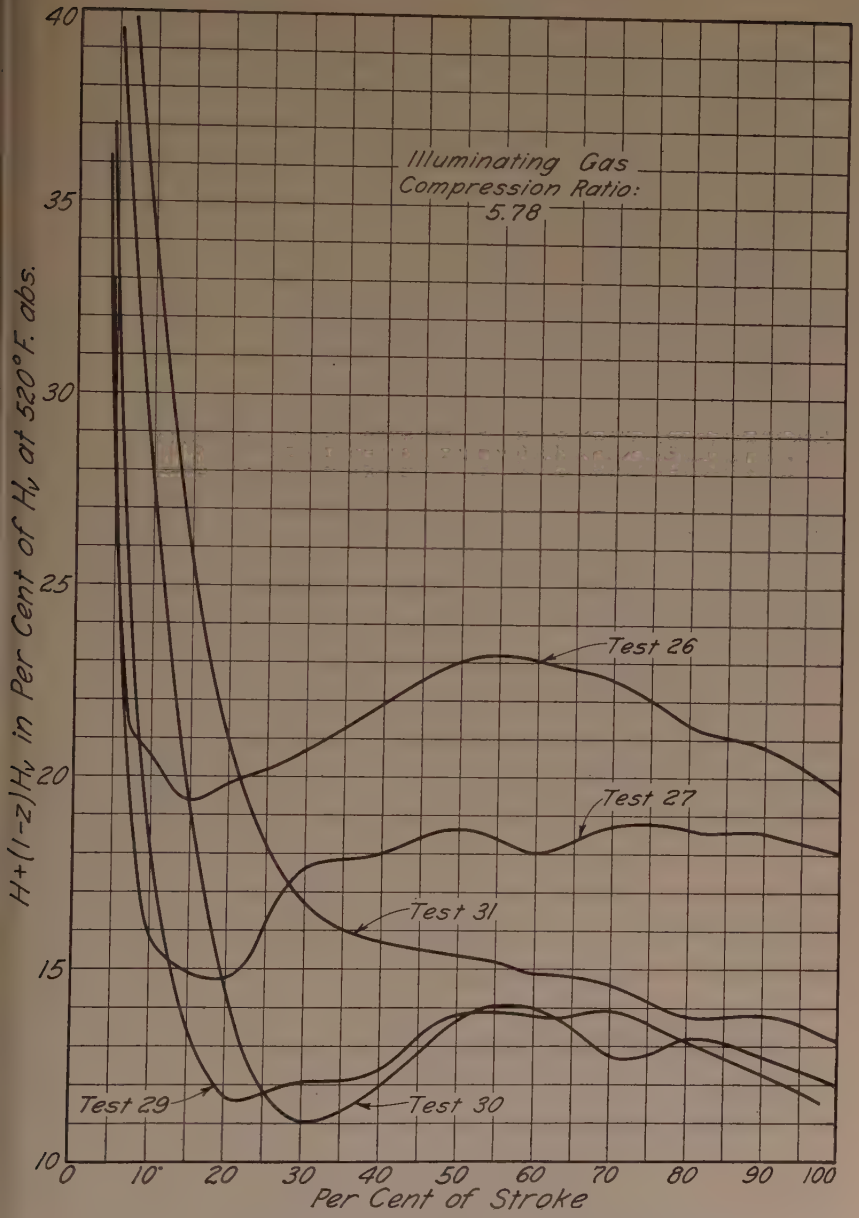

FIG. 22. CURVES OF THE FUNCTION $[H+(1-z)H_v]$ FOR TESTS AT COMPRESSION RATIO 5.78

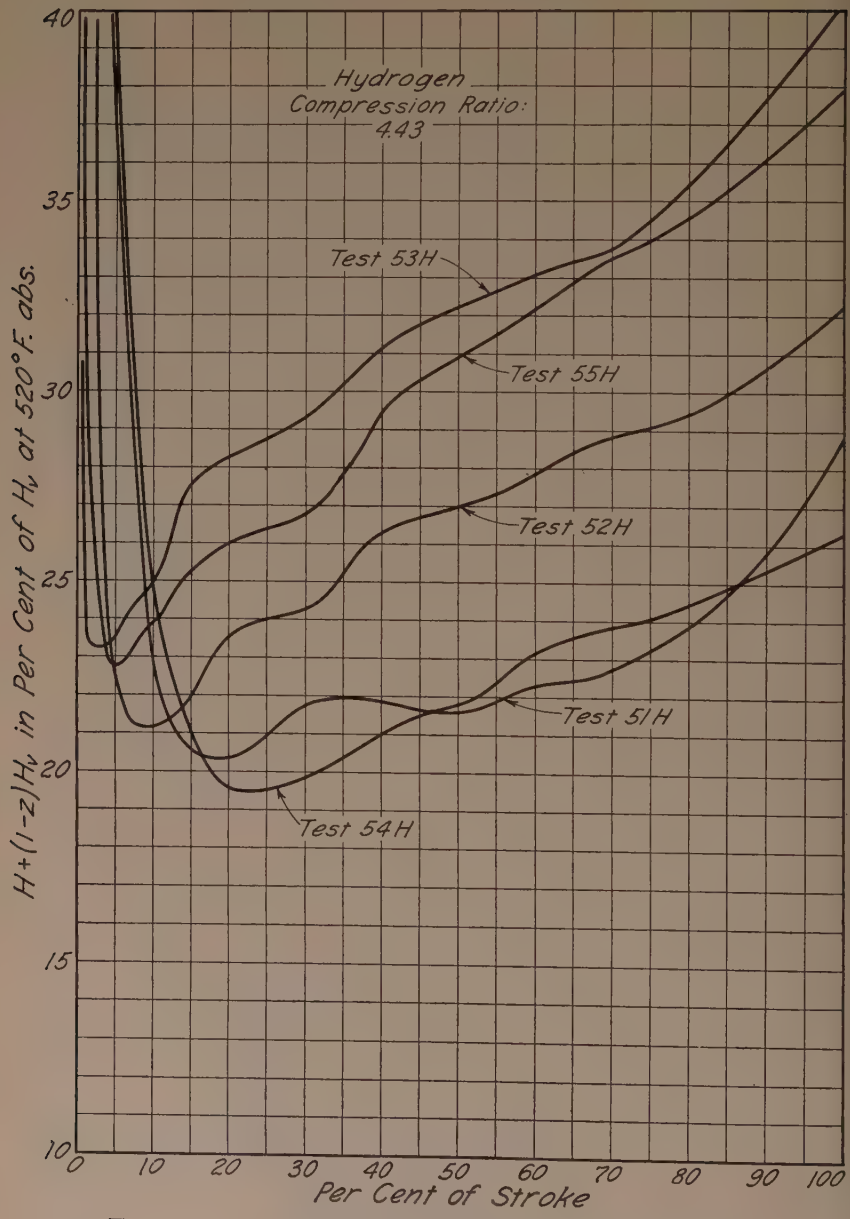

FIG. 23. CURVES OF THE FUNCTION $[H+(1-z)H_v]$ FOR TESTS WITH HYDROGEN FUEL

outlying parts (valve pockets, etc.) of the body of gas before the velocity of reaction is materially reduced. During the period of this reduced flame velocity the heat loss factor H in the expression $[H + (1 - z)H_v]$ becomes dominant and the value of the function increases. As the expansion proceeds turbulence is established, and the remaining gas mixture burns. Since the reduced temperature results in a lowered rate of heat loss, this secondary burning is sufficient to cause a reduction in the value of the function $[H + (1 - z)H_v]$. Figs. 19, 20, and 22 show the curves of the function $F = [H + (1 - z)H_v]$ for the other three series of tests. These curves lead to the same general conclusions as the ones for the 5.00 compression ratio.

55. *Effect of Mixture Ratio on Combustion Speed.*—*The* minimum points on the curves of the function $[H + (1 - z)H_v]$ may be taken as a measure of the time at which the reaction is approximately near completion. At least, the minimum point of the curve very probably represents the time at which about the same fraction of the chemical energy of reaction is liberated in each test. Hence a comparison of the times in the stroke at which the minimum points on the curves occur will probably show the influence of mixture ratio on combustion speed.

In Fig. 24 are shown the points (plotted against air-gas ratio) at which these minima occur for the various tests with illuminating gas. The tendency for the speed of reaction to decrease as the fuel mixtures become weaker is apparent. No pronounced or regular variation in the time of reaction with various compression ratios is evident, however.

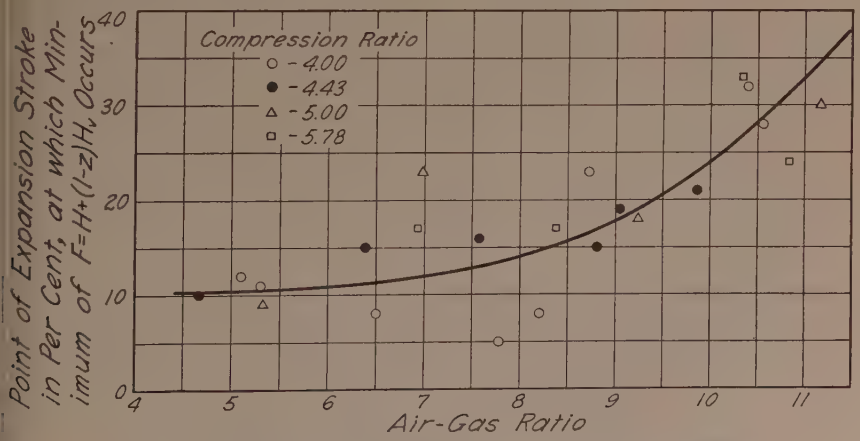

FIG. 24. EFFECT OF MIXTURE RATIO ON COMBUSTION SPEED

56. *Heat Losses.*—Figure 25 shows the relation between heat losses and air-gas ratio. The upper curves represent the actual total heat loss calculated from the weight and rise in temperature of the cooling water. The lower curves represent the value of the function $[H + (1 - z)H_v]$ at 90 per cent of the expansion stroke, or practically at the time the exhaust valve opened. The difference between the two curves represents the part of the heat lost to the exhaust, since the water jacket surrounded the exhaust valve cage.

It is evident that the heat loss as shown by both curves decreases rapidly with weaker mixtures. This decrease is the result of the lower gas temperature throughout the cycle when the weaker mixtures are used. Fig. 26 shows the variation of the calculated maximum temperature of adiabatic complete combustion with the air-gas ratio. It may be noted that the compression ratio has a comparatively small effect on the maximum temperature. This fact is further illustrated by Fig. 27, which shows the curves of actual heat loss and value of $[H + (1 - z)H_v]$ for the four compression ratios. The variation of heat loss between different compression ratios is considerably less than the variation caused by a change in the air-gas ratio. The curves of Fig. 27 show an apparent discrepancy, in that the curve for a compression ratio of 4.00 lies out of its proper position with respect to the other curves. This discrepancy may be explained by the fact that the fuels employed in each test differed in composition. A variation in the fuel will cause a considerable variation in maximum temperature, and consequently in the heat loss.

57. *Possibilities of Increased Efficiency.*—At a given compression ratio and at a given load the greatest possibility for gain in efficiency lies in the adjustment of mixture strength. In the range tested the efficiency increased from 21.90 to 29.63 per cent. In order to make possible the use of the weaker mixtures the induction system must be designed to prevent backfiring.

By the design of free passages for the inlet and exhaust valves and also by the proper selection of engine size for a given power requirement a further maximum increase from 29.63 to 34.25 per cent is possible. This 34.25 per cent efficiency is the maximum since for this case the losses due to induction and friction in the exhaust are zero.

Increases in the efficiency of the compression, explosion, and expansion process are made possible by the proper design of the combustion chamber to give the most complete combustion in a minimum time with a minimum area exposed. The design of the combustion chamber to give maximum turbulence may be more effective in increasing efficiency than the design for a minimum surface area per unit

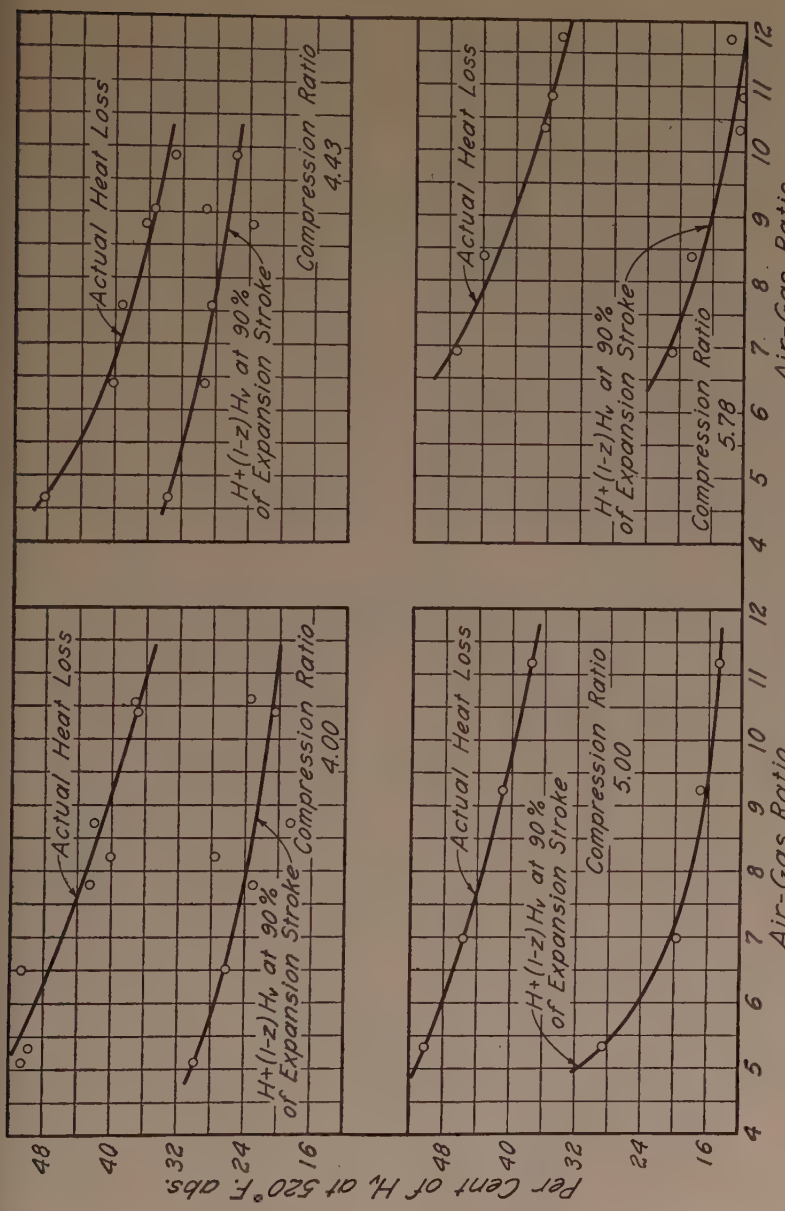

FIG. 25. HEAT LOSSES FOR TESTS AT VARIOUS COMPRESSION RATIOS

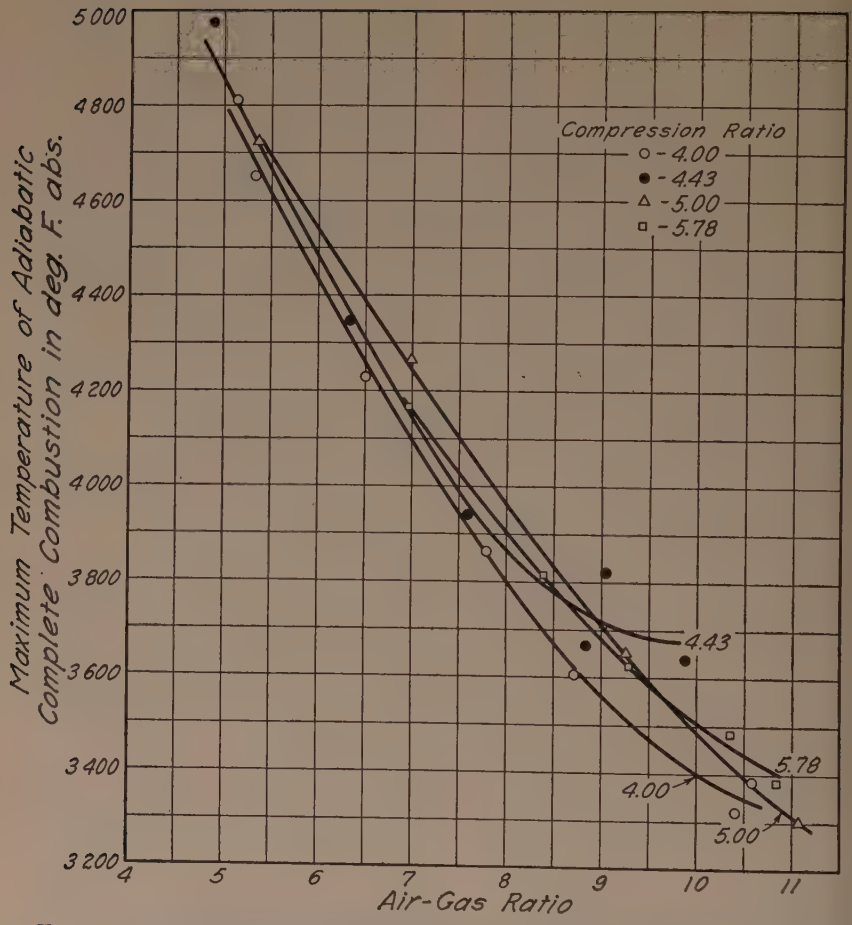

FIG. 26. EFFECT OF AIR-GAS RATIO ON CALCULATED MAXIMUM TEMPERATURE OF ADIABATIC COMPLETE COMBUSTION

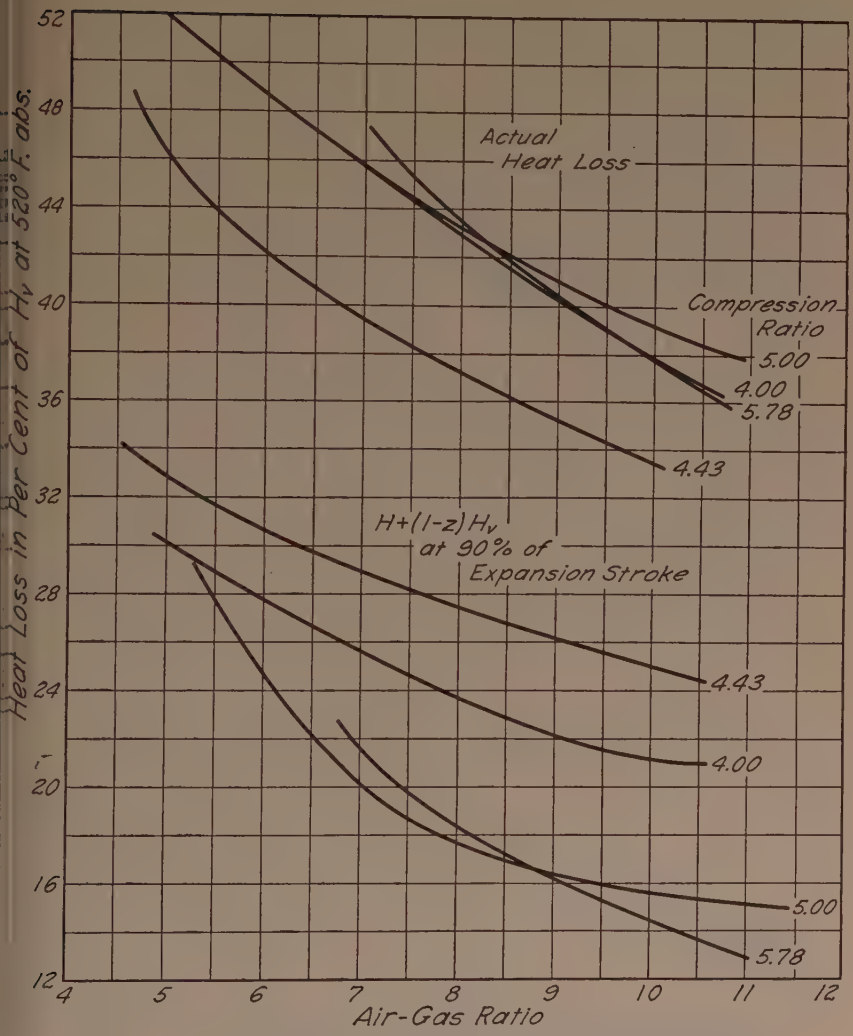

FIG. 27. HEAT LOSSES FOR TESTS WITH ILLUMINATING GAS FUEL

of volume. Valve pockets are detrimental to efficiency except in those cases where they aid in creating turbulence (L head).

No great increase in efficiency may be expected by reducing the heat lost to the jacket. For Test No. 25 an increase in efficiency from 34.25 to 42.02 per cent or only 7.77 per cent would be obtained if the explosion were instantaneous and if the heat loss of 17.8 per cent were reduced to zero. From the heat balance, Table 4, it may be seen that the major portion of any heat saved by making the cycle more nearly adiabatic is transferred to the exhaust loss and not to the work done. The increase of 7.77 per cent in efficiency made possible by the reduction of the 17.8 per cent heat loss to zero would only be obtained in the case where explosion was instantaneous. Since the explosion is not instantaneous the gain from reducing the heat loss to zero would be less than 7.77 per cent.

VI Conclusions

58. *Summary of Conclusions.*—

(1) A method has been developed for calculating the ideal adiabatic Otto cycle. This method gives results which differ from the actual cycle by amounts which can be accounted for by the various engine losses.

(2) A method has been developed for estimating the progress of the explosive reaction during the expansion stroke, and for determining the time at which the reaction is practically complete.

(3) The thermal efficiency of the engine tested at first increased with increasing air-gas ratios, finally attaining a maximum; and then decreased as the air-gas ratio increased.

(4) The thermal efficiency of the engine tested at first increased rapidly with increasing compression ratio, and tended to reach a maximum value at a compression ratio of about 6.00 to 1. If the compression ratio was increased beyond this point, the thermal efficiency decreased, and the operation of the engine became irregular.

(5) The curves of ideal thermal efficiency and indicated thermal efficiency obtained are of the same shape and lie practically parallel, substantiating the accuracy of the theoretical analysis of the cycle.

(6) The effect of dissociation with the particular fuel used was found to be very slight. The dissociation at the calculated maximum temperature in most cases was zero.

(7) The theoretical analysis gives results which agree very well with the air-standard efficiencies when the cycle is based on the operation with pure dry air.

(8) The effect of different fuels on the thermal efficiencies was found to be slight. However, the two fuels used were somewhat similar in character, and therefore no pronounced difference would be expected.

(9) It has been definitely established that the reaction in the cylinder is not complete at the instant when maximum pressure and temperature are attained.

(10) The continuation of the reaction occurring late in the stroke is sometimes caused by slow combustion, and sometimes by the fact that the gas in the valve pockets and other outlying parts of the combustion space is not ignited until late in the stroke.

(11) The general tendency of the reaction velocity is to decrease with weaker fuel mixtures. From the results of the tests herein reported, no definite effect of compression ratio on reaction velocity is evident.

(12) The heat loss during compression, explosion, and expansion decreases rapidly with increasing air-gas ratios, due to the lower gas temperatures attained with the weaker mixtures.

(13) Compression ratio has less effect on the heat loss from the gas than does the mixture ratio.

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